SEMI-ANNUAL REPORT NO. 3

DEVELOPMENT OF COMPRESSOR END SEALS STATOR INTERSTAGE SEALS, AND STATOR PIVOT SEALS IN ADVANCED AIR BREATHING PROPULSION SYSTEMS

Prepared for

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#32

PREFACE

This report describes the progress of work conducted between 1 July 1966 and 31 December 1966 by the Pratt & Whitney Aircraft Division of United Aircraft Corporation, East Hartford, Connecticut on Contract NAS3-7605, Development of Compressor End Seals, Stator Interstage Seals, and Stator Pivot Seals in Advanced Air Breathing Propulsion Systems, for the Lewis Research Center of the National Aeronautics and Space Administration.

Charles A. Knapp is Project Manager for Pratt & Whitney Aircraft for this program.

The following National Aeronautics and Space Administration personnel have been assigned to this project:

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SUMMARY

This report describes the work completed during the third six-month period of an analytical, design, and experimental program directed at developing compressor end seals, stator interstage seals, and stator pivot seals for advanced air-breathing propulsion systems.

Feasibility analyses (Tasks I and III) were discussed in the first and second Semiannual Reports (PWA-2752 and 2875) except for the analysis of the "OC" diaphragm thin strip seal, which was added to the contract by Amendment Number 2 on 15 December 1966. The basic design of this new seal concept is discussed briefly in this report under Task I.

Final design of the one side floated shoe compressor end seal has been completed. The single design point calculations of the feasibility analysis were expanded to a more nearly complete study of the variables. The seal configuration is being modified as required to meet the design objectives.

The feasibility analyses of the stator pivot seals (Task III) was completed before the start of this report period. Only a brief summary of that work is presented here.

The final designs of the stator pivot seals and the test rig are discussed under Task IV. The contractor has started hardware procurement, and expects to begin testing on 1 April 1967.

Milestone charts are presented at the end of this report.

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DEVELOPMENT OF COMPRESSOR END SEALS, STATOR INTERSTAGE SEALS, AND STATOR PIVOT SEALS IN ADVANCED AIR BREATHING PROPULSION SYSTEMS

by

A. H. McKibbin and R. M. Hawkins

ABSTRACT

The basic design concept for the "OC" diaphragm thin strip seal is discussed. The final design of the one-side floated shoe seal is discussed in some detail, including a description of component geometries, and pressure, thermal, and dynamic analyses of some components. The final designs of the vane pivot seals and test rig are also discussed.

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INTRODUCTION

High performance, modern multistage axial flow compressors built with state-of-the-art features, incorporate several air leak paths which are detrimental to compressor performance. Elimination or significant reduction of these leaks would result in a compressor of higher efficiency and possibly smaller size. Some typical areas of leak paths with estimates of percent air loss and potential effect on compressor performance are:

	% Air Loss	Effect on Compressor Efficiency
End Seal	0.6%	1.0%
Interstage Stator Seals (ten stages)	0.9%	1.0%
Vane Pivot Seals (variable stator)	0.2% per stage	0.2% per stage

Increases in compressor efficiency are traditionally sought by means of compressor geometry redesign. A few extra points in efficiency often mean the difference between a successful or an unsuccessful engine design. These increases as a result of geometry change are always very expensive and not always successful. On the other hand, the losses to efficiency as a result of air leaks are strikingly large and real gains are within reach at a relatively low cost. The gains in efficiency however, must be balanced against any detrimental effect that improved sealing may have on the engine, such as lower reliability or increased weight.

This program will provide for a research, analytical, and test program having as its goal the development of compressor end seals, stator interstage seals, and vane pivot seals which exhibit lower air leakage rates than those currently in use. This will be accomplished using components of such size, materials, and designs as to be considered applicable to compressors for engines capable of supersonic aircraft propulsion.

I. TASK I
CONCEPT FEASIBILITY ANALYSIS PROGRAMS FOR COMPRESSOR END SEALS AND STATOR INTERSTAGE SEALS

A. INTRODUCTION

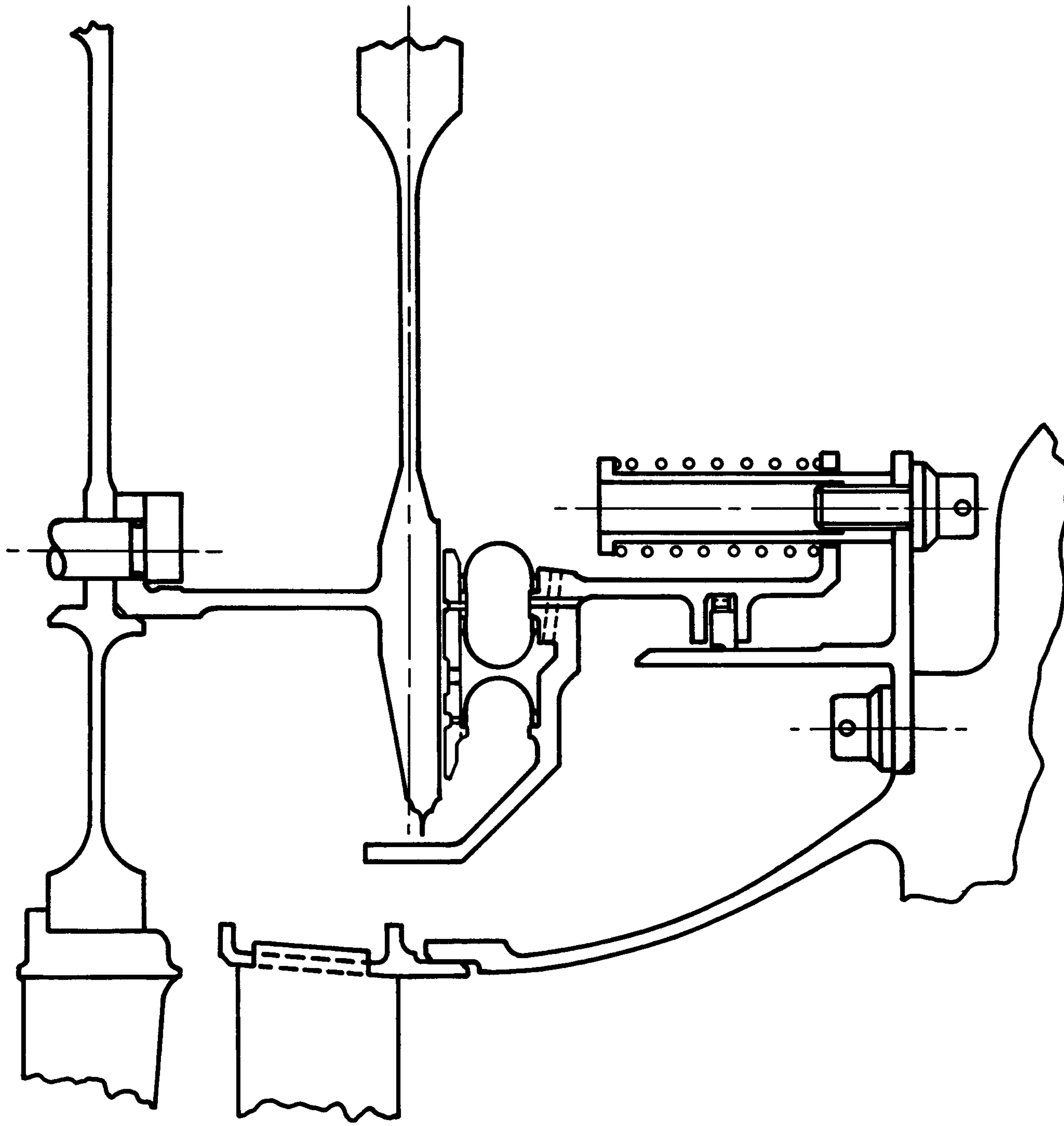
During the first 12 months of this program a feasibility analysis was conducted on compressor end seals and stator interstage seals. Work under the Task I program was completed on the one side floated shoe seal concepts, and Pratt & Whitney Aircraft submitted the feasibility designs of these seals to NASA on 19 May 1966, requesting approval to start final design under Task II. Approval was granted in a letter from NASA dated 31 May 1966.

The results of the feasibility analysis indicated that the two side floated shoe seal (a radial seal) was also worthy of final design and manufacture. Recommendation of this type of seal was held in abeyance because the similarity of the two floated shoe seal designs would leave the program without a backup of radically different concept. Contract Amendment No. 2, effective 15 December 1966, was received from NASA for the feasibility analysis of an "OC" diaphragm thin strip seal design. As a result of this new amendment, the period of performance of Contract NAS3-7605 has been extended from 26 1/2 months to 43 months. A more detailed discussion of the feasibility analyses and their results is contained in the first two Semiannual Reports (PWA-2752 and PWA-2875).

Request for approval of the second seal configuration is being withheld pending the outcome of the work accomplished under Contract Amendment No. 2

B. "OC" DIAPHRAGM THIN STRIP SEAL

A feasibility analysis has been initiated for the "OC" Diaphragm Thin Strip Seal for application in compressor end seal and stator interstage seal locations in advanced air-breathing propulsion systems. Figure 1 illustrates the seal concept. Before starting the detailed feasibility analysis, the contractor will conduct a preliminary analysis and verification of the basic seal design concept. Preliminary analysis will include studies of the Rayleigh step and spiral groove primary seal faces. Selection of the most promising seal face will permit further analyses of dynamic seal response, leakage, and other seal characteristics.



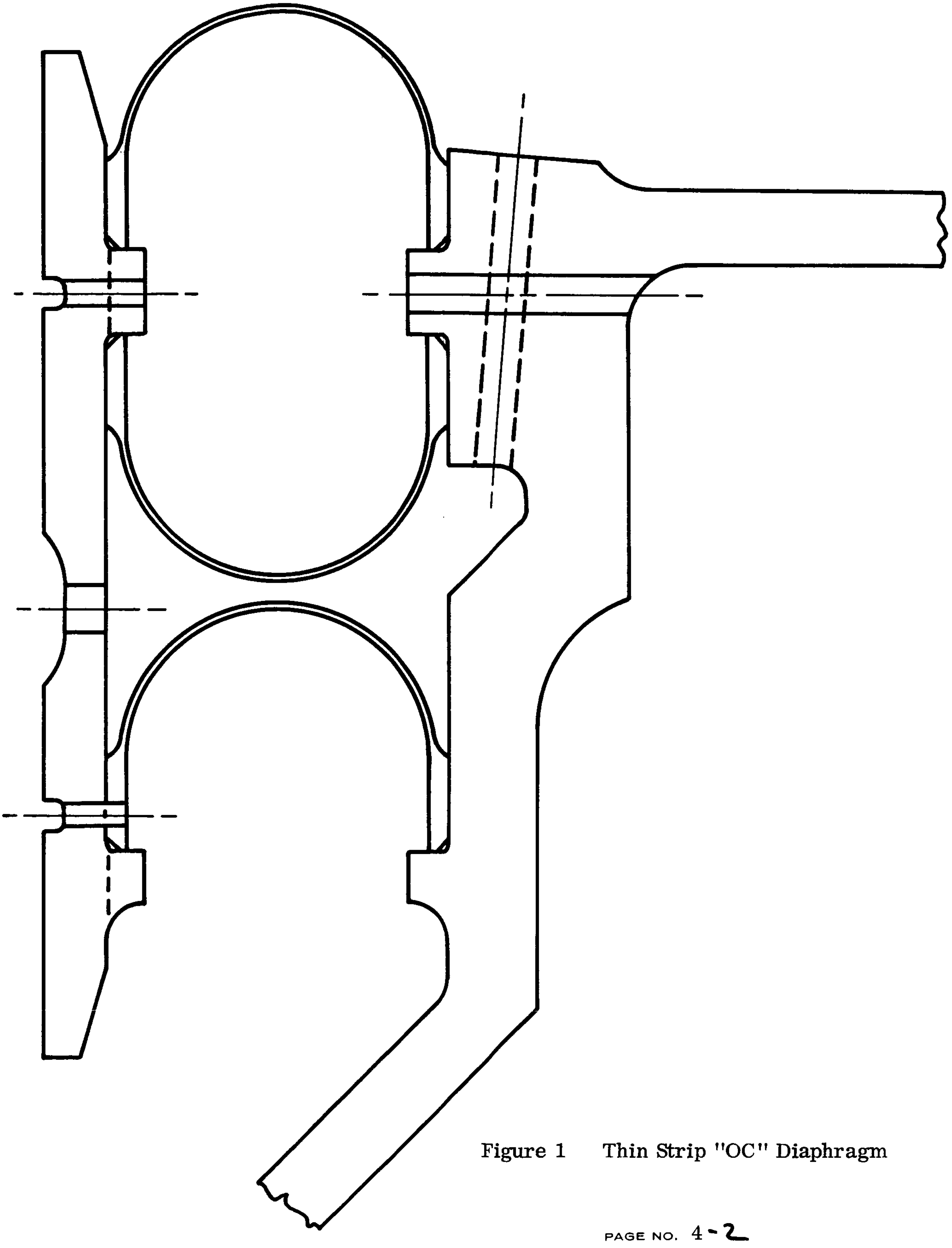


Figure 1 Thin Strip "OC" Diaphragm

II. TASK II COMPRESSOR END SEAL AND STATOR INTERSTAGE SEAL EXPERIMENTAL EVALUATION

A. INTRODUCTION

This phase of the program provides for final design and procurement of compressor end seals and stator interstage seals, design and fabrication of a test rig, and experimental evaluation of the compressor seals.

The final design of the four compressor seal concepts selected for experimental evaluation includes all calculations, material determinations, analyses, and drawings necessary for seal optimization, procurement, and experimental evaluation. A test rig will be designed and fabricated to evaluate the selected compressor end seals and stator interstage seals under simulated compressor operating conditions. The test apparatus will simulate the last stages of a full scale compressor including supporting members and bearing system in order to faithfully duplicate structural flexibility and thermal gradients.

The compressor end seals and stator interstage seals will be calibrated in incremental steps at room temperature static conditions, room temperature dynamic conditions, and subsequently over the full speed, pressure, and temperature operating range. Finally, the seals will be subjected to endurance testing.

B. DESIGN OF THE ONE SIDE FLOATED SHOE COMPRESSOR END SEAL

Final design of one of the four compressor seals has been completed. A thorough review of the one side floated shoe end seal was undertaken with the following objectives:

- Define the most effective approaches to light-weight seal design so that realistic engine weight penalty estimates are available.
- Ensure a minimum engine starting torque.
- Determine the tolerances required for all critical seal dimensions, considering the full spectrum of operating conditions.

Because studies of the above items completed in Task I were preliminary and could lead to inaccurate judgement of the merits of the design, the single design point calculations of the feasibility analysis were expanded to a more complete study of the variables. The seal configurations are being adjusted as required to meet the design objectives.

The one side floated shoe seal is a face seal consisting of a ring of segments acting against a rotating surface attached to the compressor rotor. Figure 2 is a schematic drawing of the seal. The rotating surface is flat, and the leakage flows radially inward through the seal. The primary seal is between the stationary ring of shoes and the rotating face. The secondary seals are between the shoes and the carrier ring, and between the carrier ring and mounting ring.

Most of the weight of the seal assembly is concentrated in three parts: the carrier, the shoes, and the rotating plate. The shoe design weight is controlled by force balancing and thermal considerations. It has been kept low to ensure good tracking characteristics. Carrier and rotor design weights, however, are governed primarily by stress and deflection limits which must be established in the final design analysis. Attention must also be given to the carrier and rotor temperature distribution in order to avoid severe angular distortion from axial thermal gradients. Changes to the original one side floated shoe seal configuration have led to significant decreases in seal weight and starting torque.

It is anticipated that the end seal design will be completed by 31 January 1967 after which NASA approval will be requested and detail drawings will be started. Final design of the one side floated shoe interstage seal will be started in February.

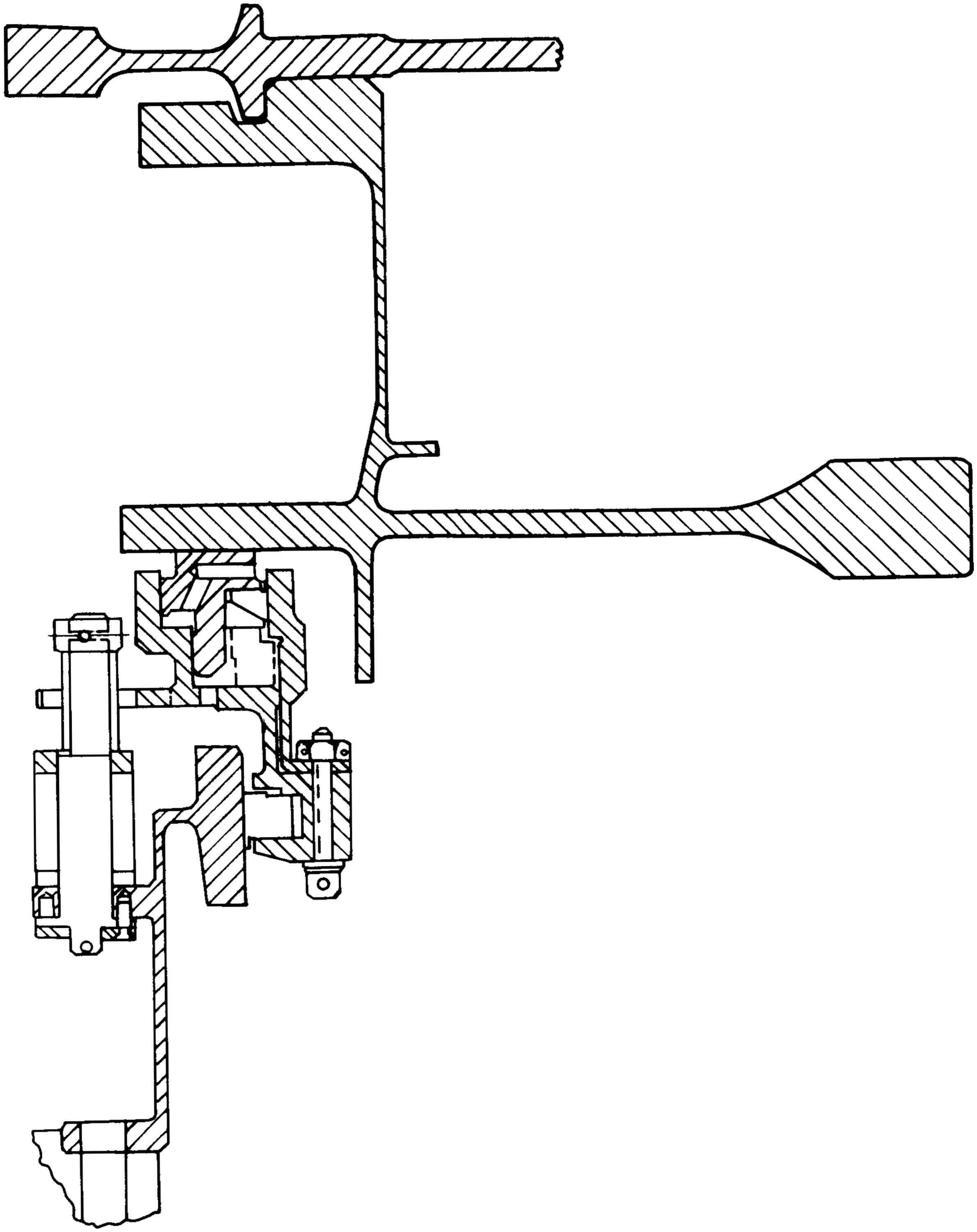


Figure 2 Schematic of One-Side Floated Shoe Seal

C. PRIMARY SEAL ANALYSIS

During the third semiannual report period, an error was discovered in the computer program which was used to analyze the primary seal interface geometry. That program (discussed in Appendix A to the Second Semiannual Report, PWA-2875) contained a minor error in flow continuity relationships. The error was not significant enough to affect the conclusions regarding seal performance or the overall seal configuration. A listing of the corrected program is shown in the Appendix to this report. A comparison of the original calculations and the corrected calculations is shown in Table I.

TABLE I

PRIMARY SEAL CHARACTERISTICS AT THE ALTITUDE CRUISE DESIGN POINT

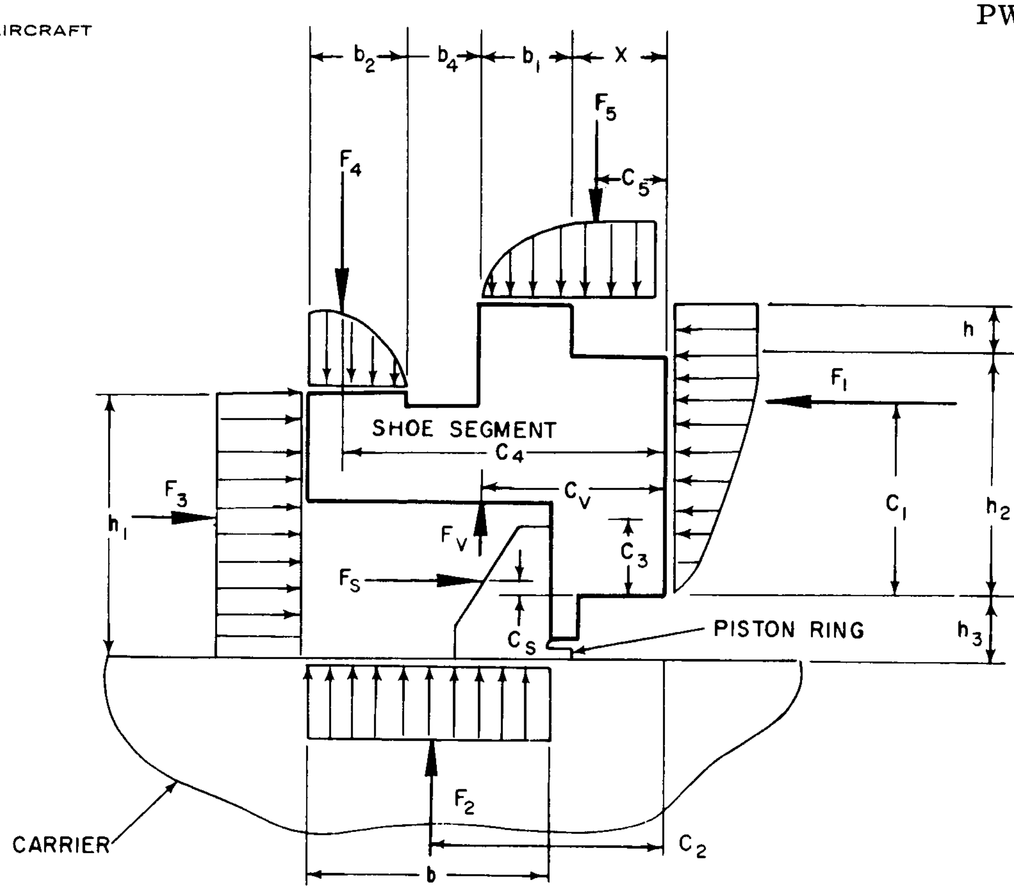
	h	$\overline{W}$	$\overline{K}$	$\overline{Q}$	$\overline{X}$
Original Calculation	0.001	0.802	0.116	37.3	0.437
Corrected Calculation	0.001	0.8245	0.158	39.3	0.435

The favorable increases in load capacity ($\overline{W}$) and stiffness ($\overline{K}$) are attributable to a more significant hydrodynamic effect in the corrected computer program. Changes to certain seal dimensions are required in order to maintain the intended 0.001 inch film thickness at the cruise operating condition.

The error in the original program was made when the mathematical analysis was converted to Fortran computer language. The bulk of the primary seal data obtained during this report period was generated with the defective program, and will not be discussed in detail in this report. The data included the results of studies of the primary seal under simulated starting conditions, and evaluations of the effects of seal wear and of non-uniform film thickness caused by distortion of the seal parts.

D. SHOE GEOMETRY

A digital computer was programmed to solve the moment and balance equations for the primary shoe. Figure 3 shows diagrammatically the forces acting on the shoe and lists the equations used for balancing. Figure 4 is a listing of the computer program used in solving the equations. Because of a slight gas-force imbalance on the carrier, the spring force on the shoe is a function of pressure, and varies accordingly. The tabulated results of the balancing analysis (Figure 3) give two different values: one for cruise and one for take-off. The difference is due to different values of dimensionless load capacity ($\overline{W}$) and center of pressure ($\overline{X}$) at these two conditions. Since the primary seal can be physically designed with only one set of dimensions (for instance those corresponding to a cruise condition), deviations from the design point film thickness or an unbalanced moment will produce a small rotation of the shoe at any other operating condition. Variation of dimensions due to tolerances on the shoe or carrier will have the same effect.



FORCE BALANCE — ℄ —

HORIZONTAL — $F_1 = F_S + F_3$

VERTICAL — $F_2 = F_5 + F_4 - F_V$

MOMENT BALANCE

$$F_1 C_1 + F_5 C_5 + F_4 C_4 = F_2 C_2 + F_S C_S + F_3 C_3 + F_V C_V$$

RESULTS

$h_2 = 0.500$ $X = 0.160$

$b_1 = b_2 = 0.230$ $Y = 0.100$

$b_3 = 0.234$ $b_4 = 0.130$

CRUISE:

$\Delta P = 80$ PSI $F_S = 0.8737$ LB/IN. $F_V = 0.5242$ LB/IN.

$\overline{W}_{PRIMARY} = 0.8245$ $\overline{W}_{SEC} = 0.790$

$\overline{X}_{PRIMARY} = 0.435$ $\overline{X}_{SEC} = 0.443$

$h_1 = 0.5013$ IN. $h_3 = 0.0658$ IN. $b = 0.5168$ IN.

TAKE-OFF:

$\Delta P = 150$ PSI $F_S = 1.3713$ LB/IN. $F_V = 0.8228$ LB/IN.

$\overline{W}_{PRIMARY} = 0.8090$ $\overline{W}_{SEC} = 0.810$

$\overline{X}_{PRIMARY} = 0.436$ $\overline{X}_{SEC} = 0.446$

$h_1 = 0.4954$ IN. $h_3 = 0.0607$ IN. $b = 0.5271$ IN.

Figure 3 Force and Moment Balance of One-Side Floated Shoe Seal

```
JCB  P135    02  9  1DH  6240  NICOLICH,  PAT                       01.51.20
CPTICN   LINK
PHASE SPIRAL,*
EXEC       FCRTRAN

                               DISK OPERATING SYSTEM/360 FORTRAN   360N-FO-451 10

           WRITE(6,10)
        10 FCRMAT(56H COMPRESSOR SEAL  -  ONE SIDE FLOATED AND INVERTED SHOE
          113X,17H** P. NICOLICH **///)
        11 REAC(5,12) ICASE,X,Y,C
        12 FCRMAT(I1,F9.4,F10.4,F10.4)
           IF(ICASE)14,14,15
        15 REAC(5,13) WBP,WB1,WB2,WB3,XBP,XB1,XB2,XB3
           REAC(5,13)B1,B2,B3,B4,H2,HS,HV,DP
        13 FCRMAT(8E1C.3)
           F=.583*CP+25.
           FS=F/82.
           FV=FS*.6
           WRITE(6,16) ICASE
        16 FCRMAT(10X,12H-----*****  , 9HCASE NO.=I2,12H  *****----- /)
           WRITE(6,17)WBP,WB1,WB2,WB3,XBP,XB1,XB2,XB3,FS,HS
        17 FCRMAT(5H WBP=F7.4,8H    WB1=F7.4,8H    WB2=F7.4,8H    WB3=F7.4,/
          15H XBP=F7.4,8H    XB1=F7.4,8H    XB2=F7.4,8H    XB3=F7.4,8H     FS
          2=F7.4,8H      HS=F7.4/)
           WRITE(6,18) B1,B2,B3,B4,H2,X,Y,C,FV,HV,ICASE
        18 FCRMAT(5H  B1=F7.4,8H     B2=F7.4,8H     B3=F7.4,8H     B4=F7.4,/
          15H  H2=F7.4,8H     X=F7.4,8H     Y=F7.4,8H     C=F7.4,8H     FV
          2=F7.4,8H      HV=F7.4,
          3      //12H O U T P U T, 7X ,9HCASE NO.=I2,/)
           B=B2*WB2+B1*WB1+X-FV/DP
           H1=H2*WBP+Y-FS/DP
           R1=-WBP*H2*H2*(1.-XBP)
           R2=-Y*(H2+.5*Y)-.5*X*X
           R3=-WB1*B1*(X+XB1*B1)
           R4=-WB2*B2*(X+B1+B4+B2-XB2*B2)
           R5=B*(.5*B+X+B1+B2-B+B4)
           R6=FV*HV/CP
           R7=FS*HS/CP
           R=R1+R2+R3+R4+R5+R6+R7+H1*H1/2.0
           H3=R/H1
           SH=H3+H2+Y-H1
           E=X+B1+B2+B4-B-B3*(1-WB3)
           WRITE(6,19) B,H1,H3,DP,SH,E
        19 FCRMAT(5H   B=F7.4,8H     H1=F7.4,8H     H3=F7.4,8H     DP=F7.3,
          18H      SH=F7.4,8H      E=F7.4//)
           GC TC 11
        14 CALL EXIT
           ENC
```

Figure 4   Fortran Computer Program for Shoe Balancing

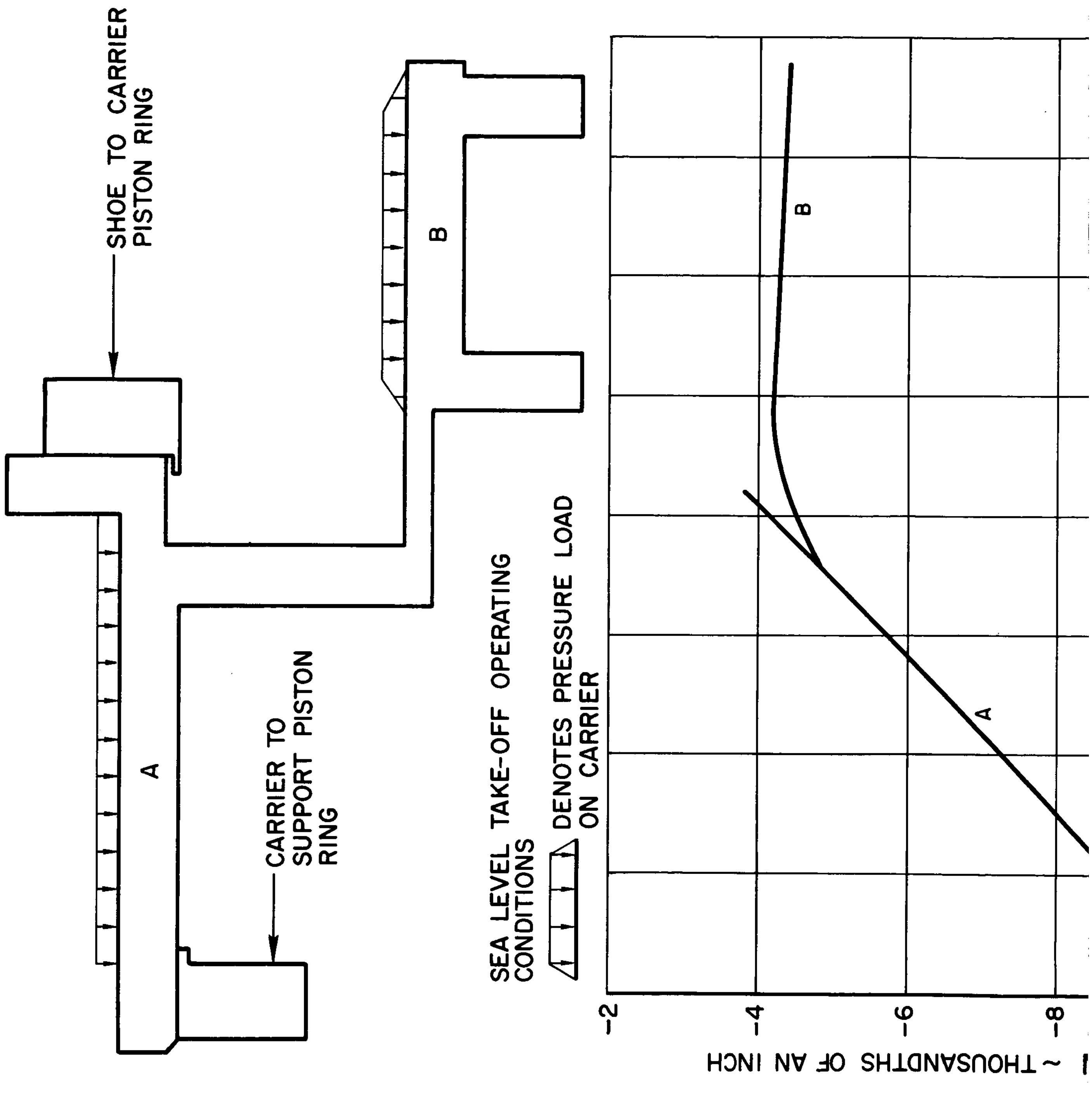
E. CARRIER GEOMETRY

Design studies of the seal carrier have included a final analysis of deflections, study of vibration characteristics and spring rates, and a final heat transfer analysis. Analysis to date has indicated that the structural design of the carrier and springs will achieve a satisfactory combination of low weight and low resonant frequency.

Initially, two basic carrier designs were investigated. One configuration placed the carrier-to-support piston ring in the seal support, as in conventional face seal practice. This design is shown in Figures 5 and 6. It was eliminated because the angular deflections far exceed the 0.001 radian limit which must be placed on the secondary seal surfaces. The second configuration places the piston ring groove in the carrier, as shown in Figure 7. This configuration was chosen for further development because of its relatively small deflections and because it is not sensitive to axial travel of the carrier.

Design development of the second configuration discussed above has resulted in a new carrier, shown in Figure 8. Figure 8 also shows the deflection and slope of the carrier when the shoe is at its mean operating position on the carrier. Extreme shoe positions cause minor changes in the pressure load, but under no condition does the carrier slope exceed 0.001 radians at the secondary seal locations. Deflections caused by temperature gradients are shown in Figure 9.

A study of the vibration characteristics of the carrier-shoe-spring system has evaluated several alternatives in the choice of carrier mass and spring stiffness. Figures 10 through 12 show the variation of shoe amplitude and carrier amplitude with rotor speed for three combinations of mass and stiffness. Figure 13 shows the effect of each mass-stiffness combination on primary film thickness. The combination of a heavy carrier and stiff spring is that of the feasibility analysis, and produces a resonance below engine idle speed. The combination of a light carrier and a stiff spring produces a resonance above the maximum engine speed, but would require extremely stiff springs if disturbances to the primary film thickness in the engine operating range are to compare favorably with those of the first case. The combination of a light carrier and flexible spring returns the resonance to a speed below engine idle, and narrows the speed range over which serious disturbances to the primary film occur. All curves describe the response of an undamped system. The light carrier and flexible spring combination has been selected for the final configuration.


SHOE TO CARRIER PISTON RING
B
A
CARRIER TO SUPPORT PISTON RING
SEA LEVEL TAKE-OFF OPERATING CONDITIONS
DENOTES PRESSURE LOAD ON CARRIER
-2
-4
-6
-8
~ THOUSANDTHS OF AN INCH

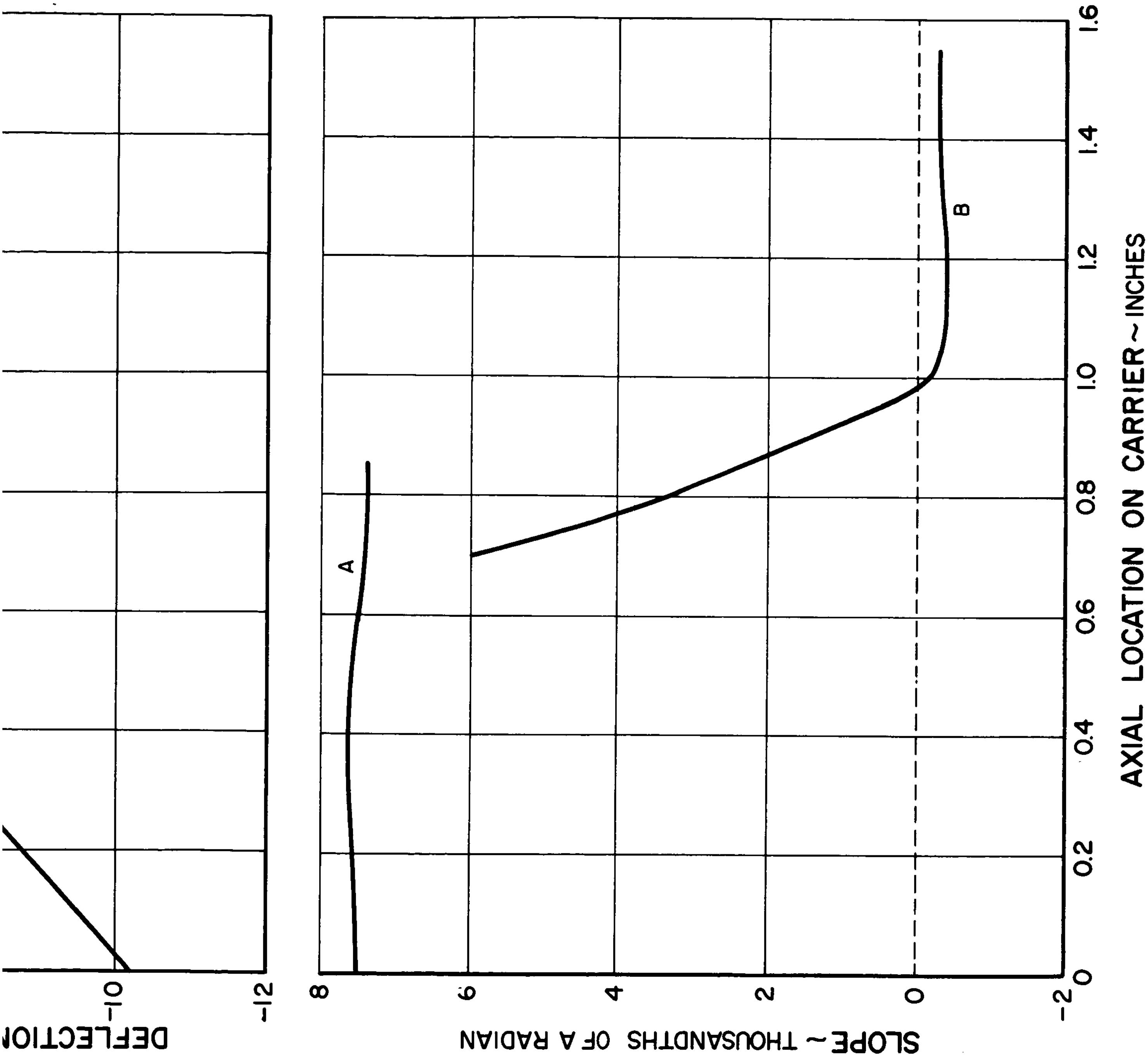
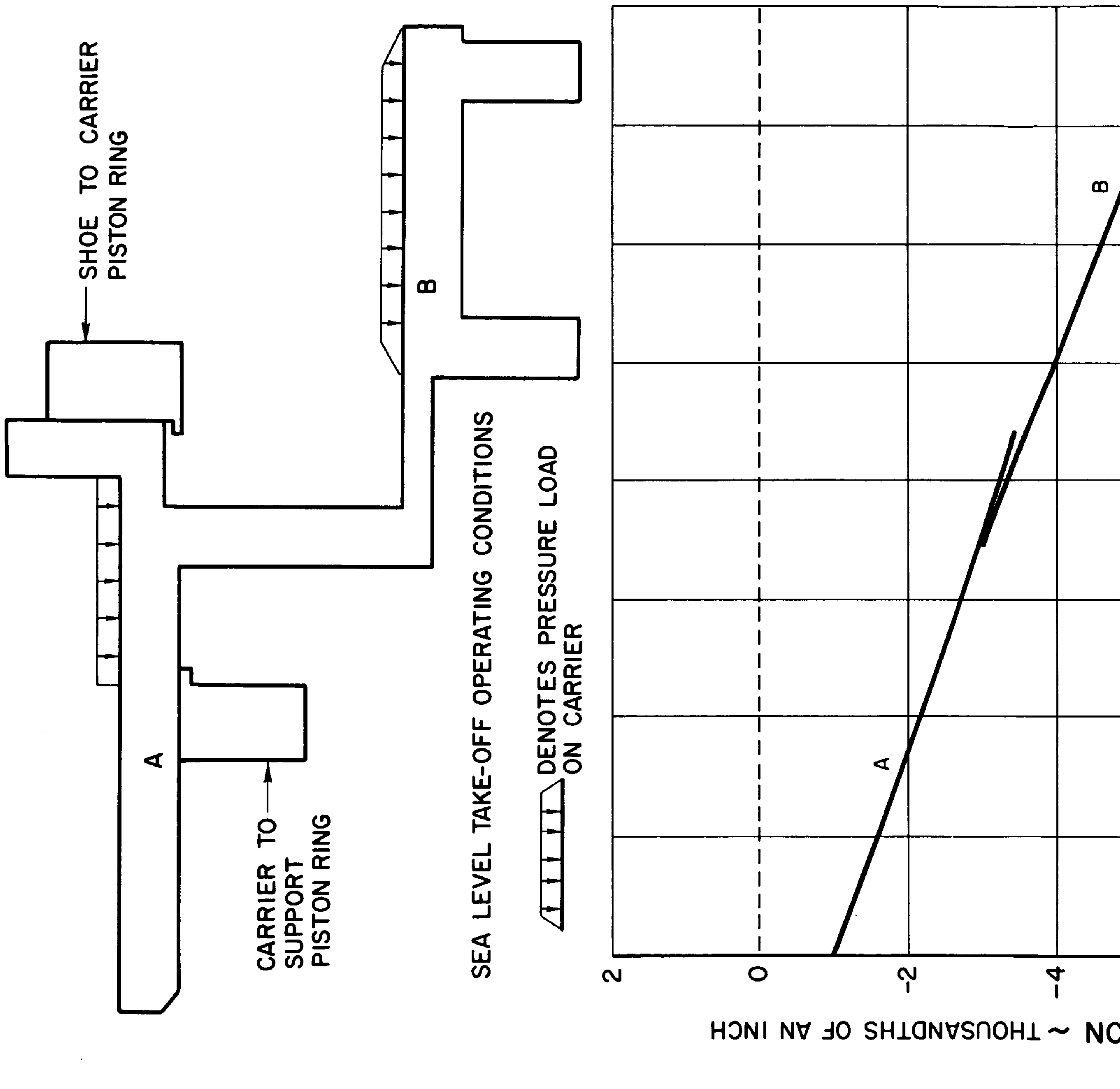


Figure 5 Seal Carrier Deflection with Piston Ring Groove in Seal Support and Carrier at Outer Extreme of the 0.4" Axial Travel


SHOE TO CARRIER
PISTON RING
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DENOTES PRESSURE LOAD
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0
-2
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ON ~ THOUSANDTHS OF AN INCH

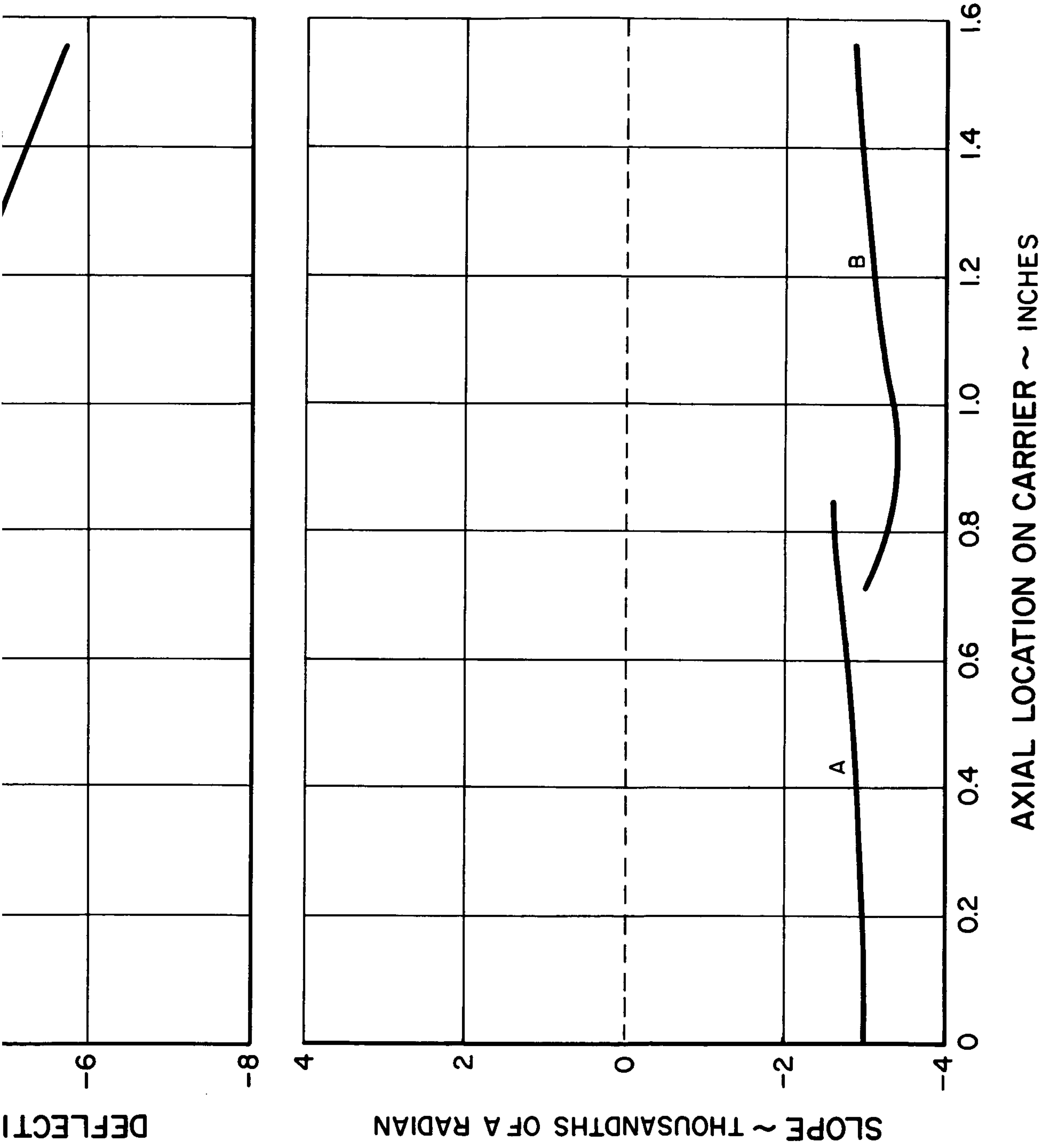
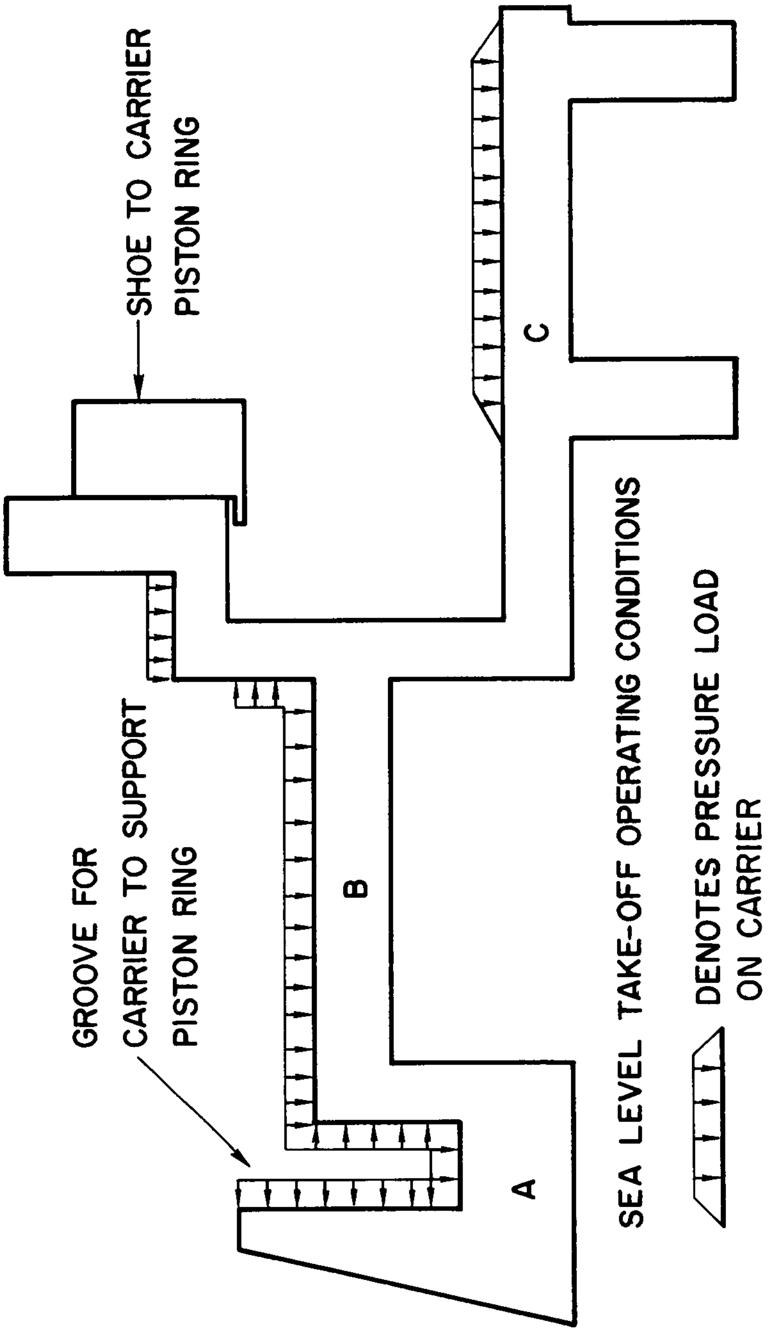


Figure 6 Seal Carrier Deflection with Piston Ring Groove in Seal Support and Carrier at Inner Extreme of the 0.4" Axial Travel


SHOE TO CARRIER
PISTON RING
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GROOVE FOR
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ON CARRIER

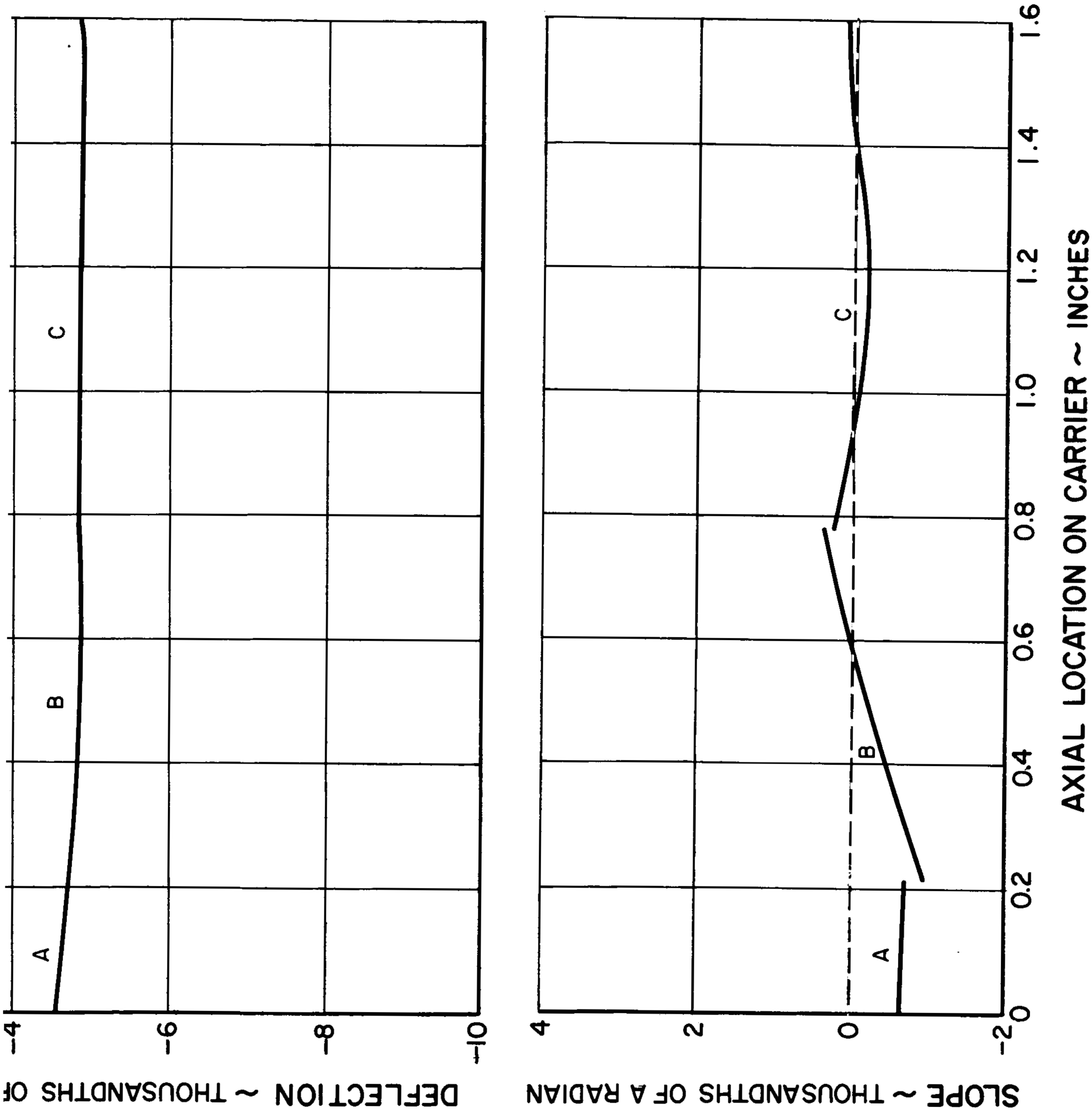
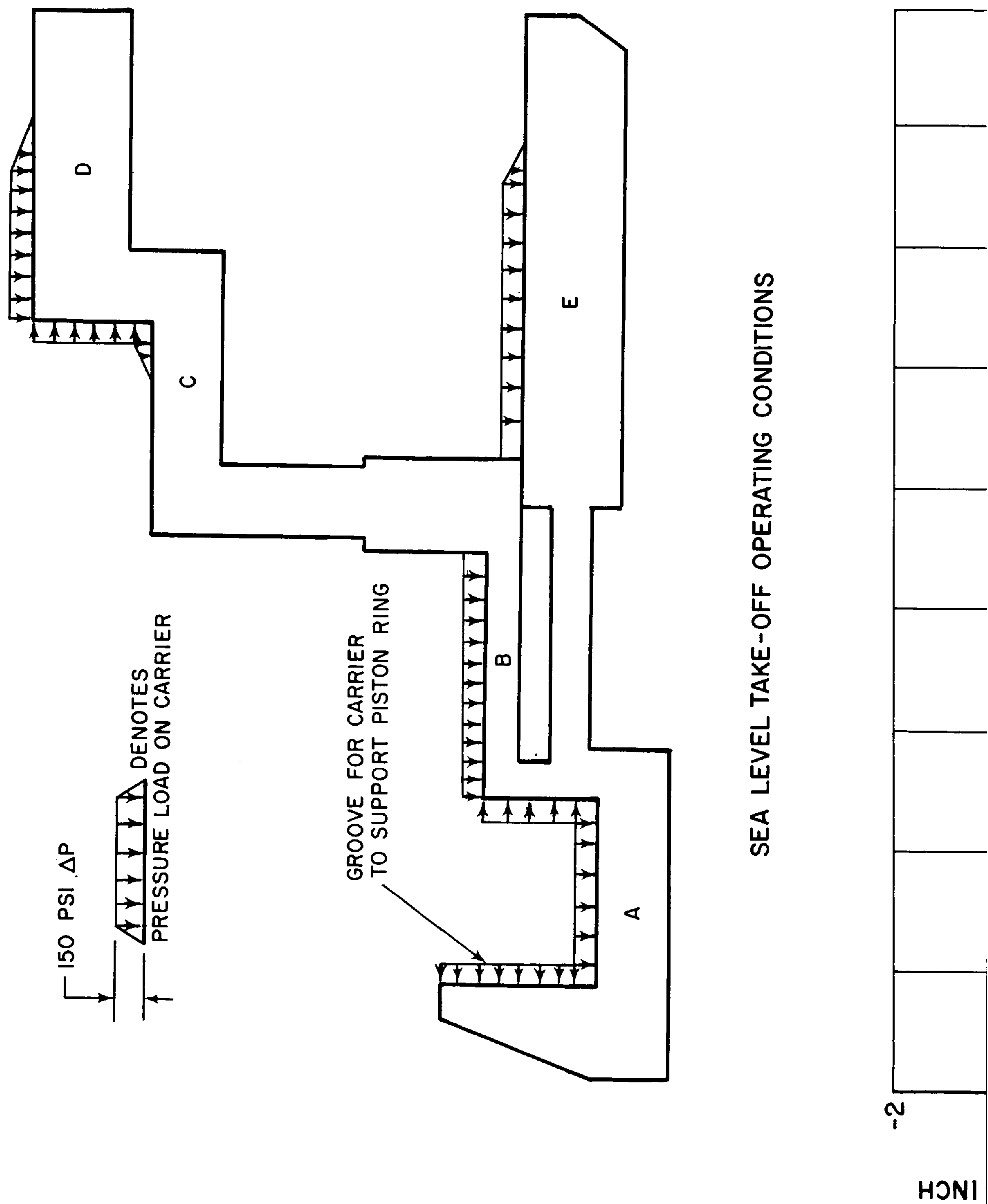


Figure 7   Seal Carrier Deflection with Piston Ring Groove in Carrier and Carrier at any Position



SEA LEVEL TAKE-OFF OPERATING CONDITIONS

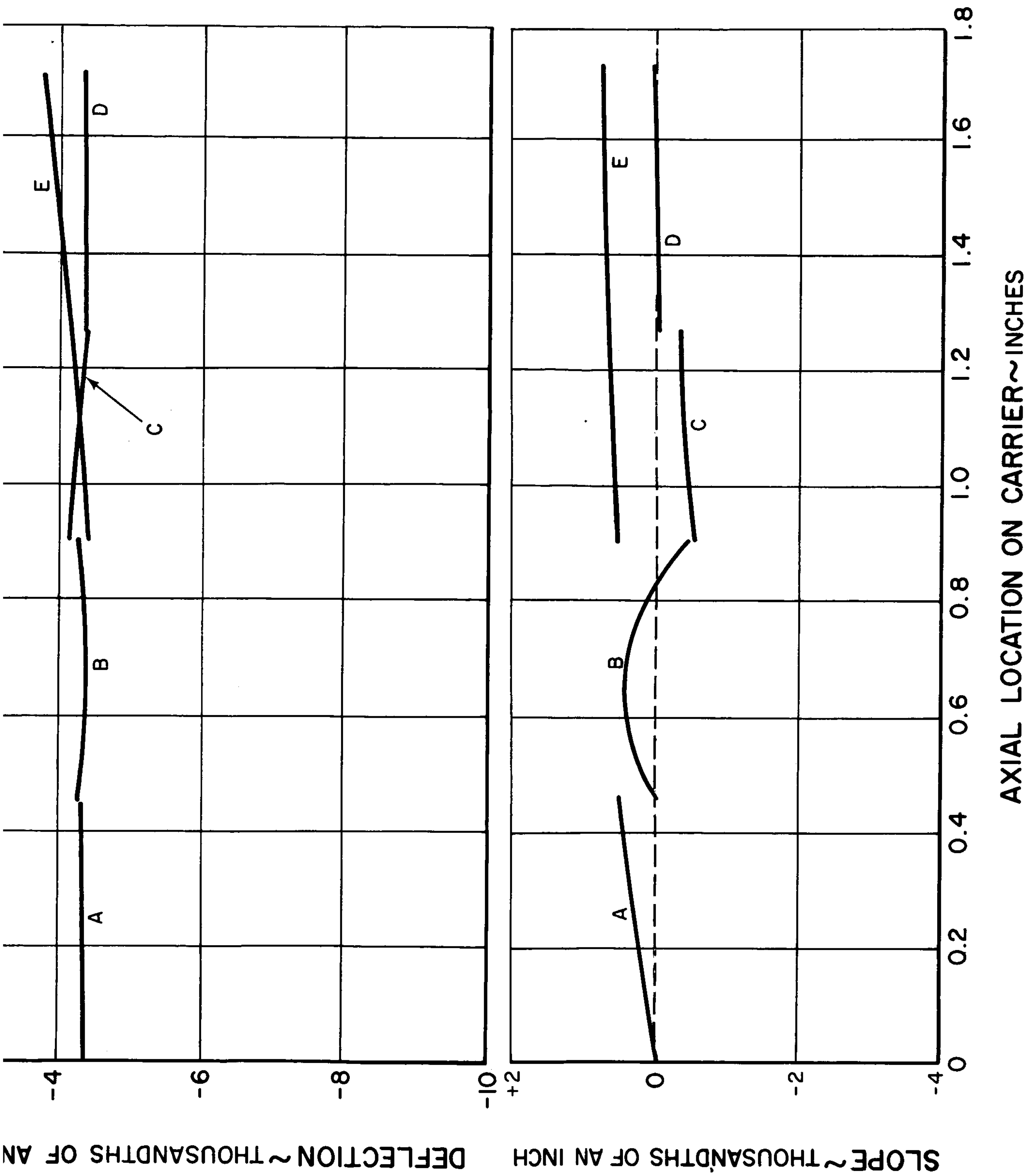
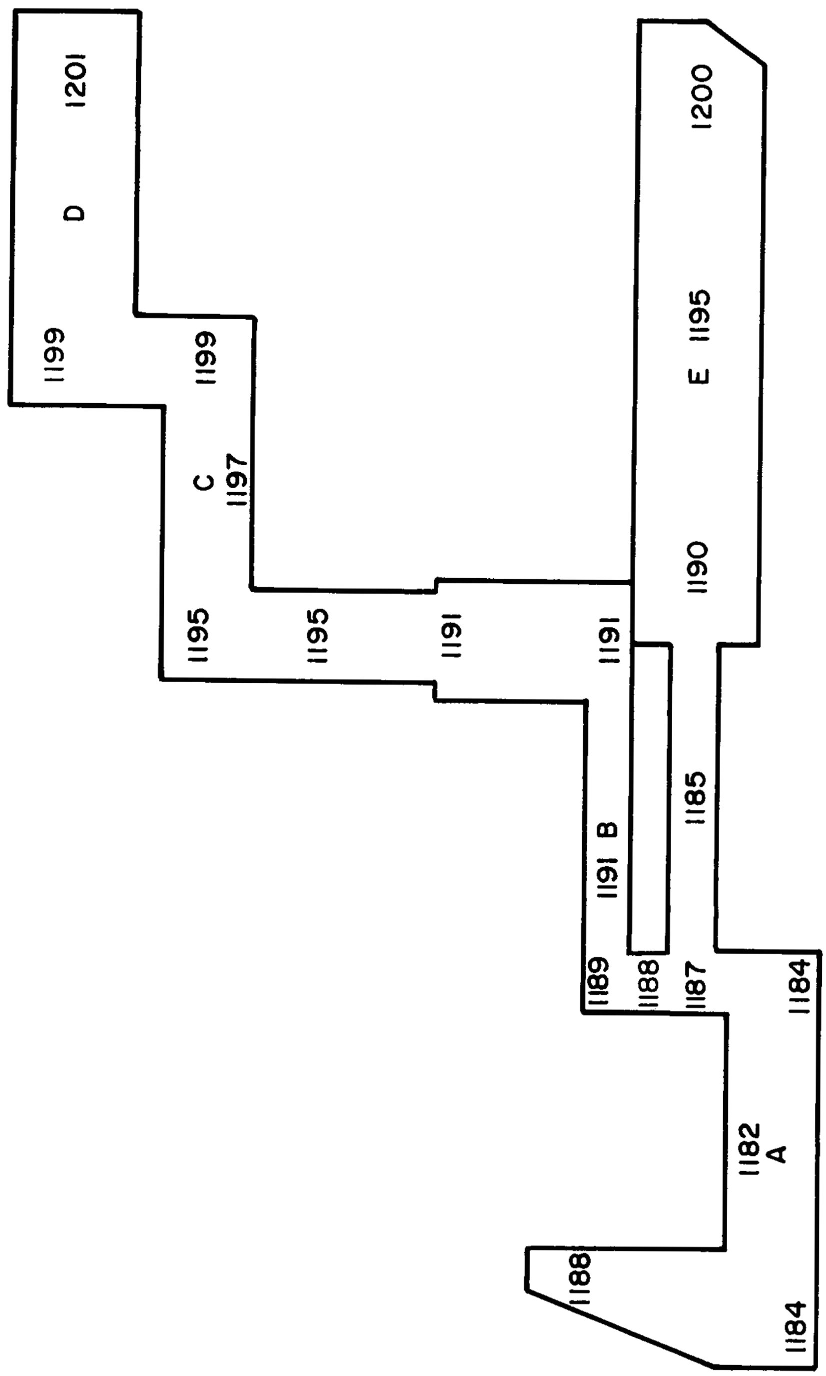


Figure 8 Seal Carrier Deflection and Slope with Shoe in Mean Operating Position, Final Configuration



CRUISE OPERATING CONDITIONS

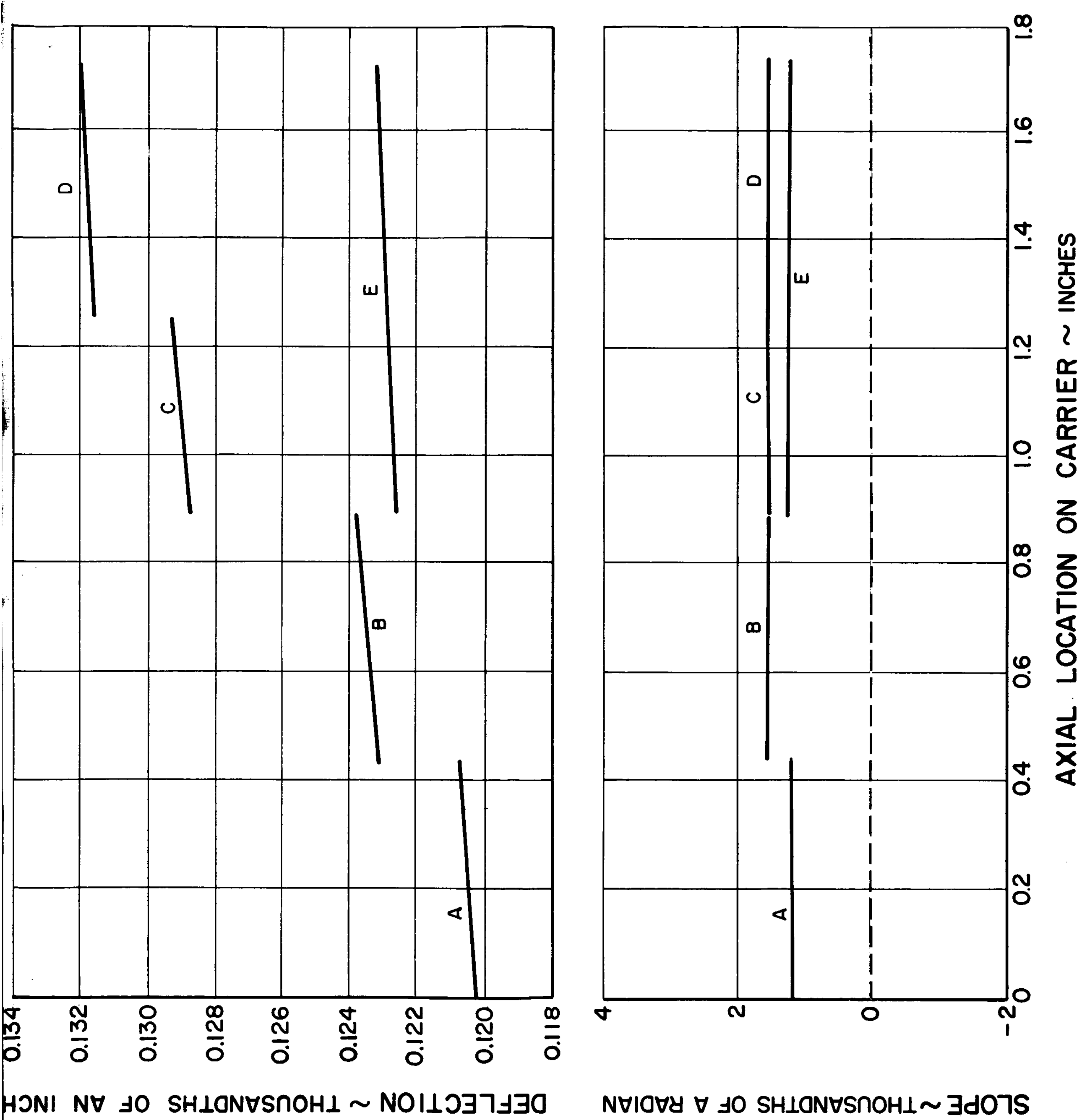


Figure 9 Seal Carrier Deflection Caused by Temperature Gradients

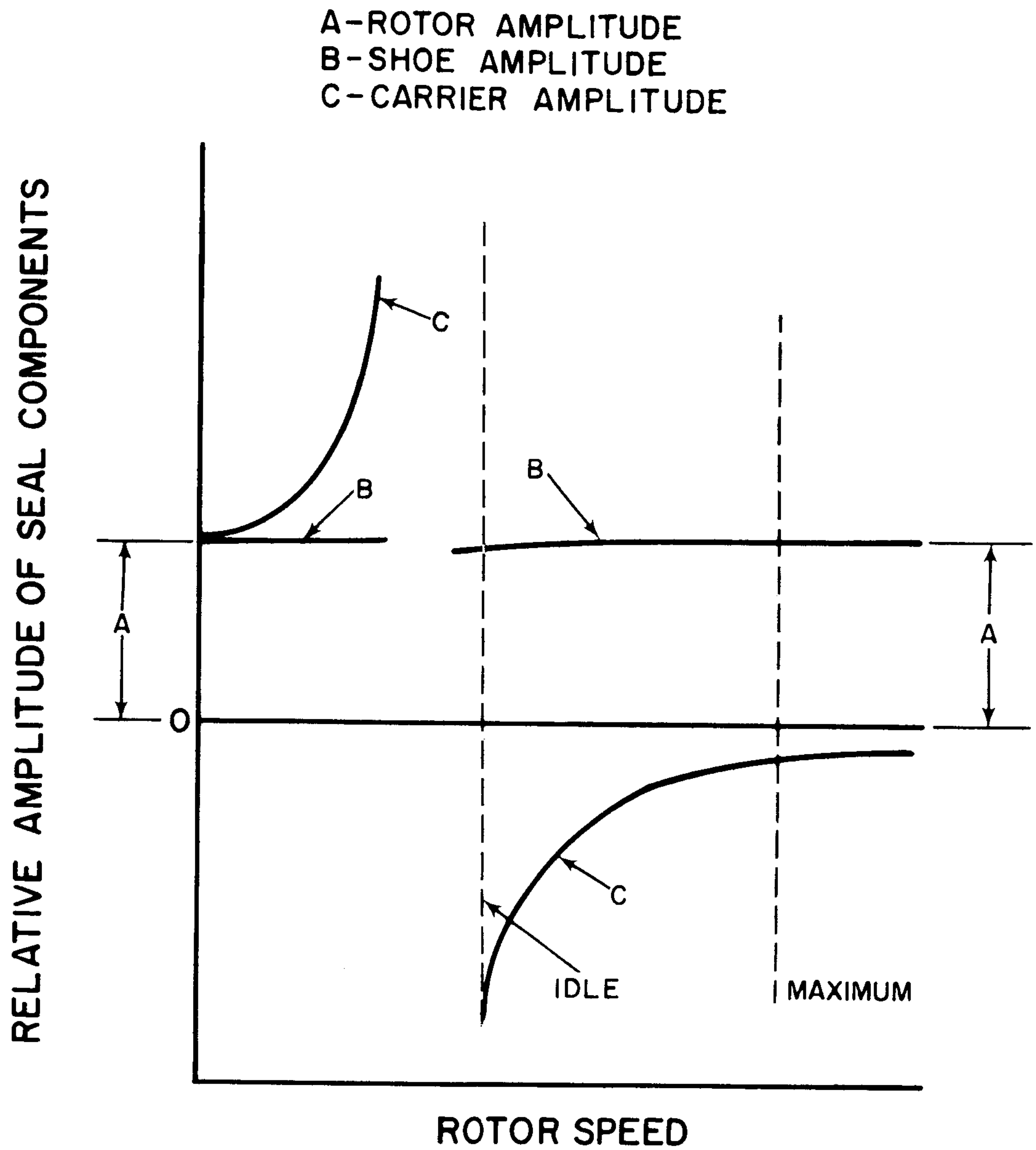


Figure 10 Variation of One-Side Floated Shoe Seal Vibration Amplitude with Rotor Speed for a Light Carrier and Flexible Wave Spring

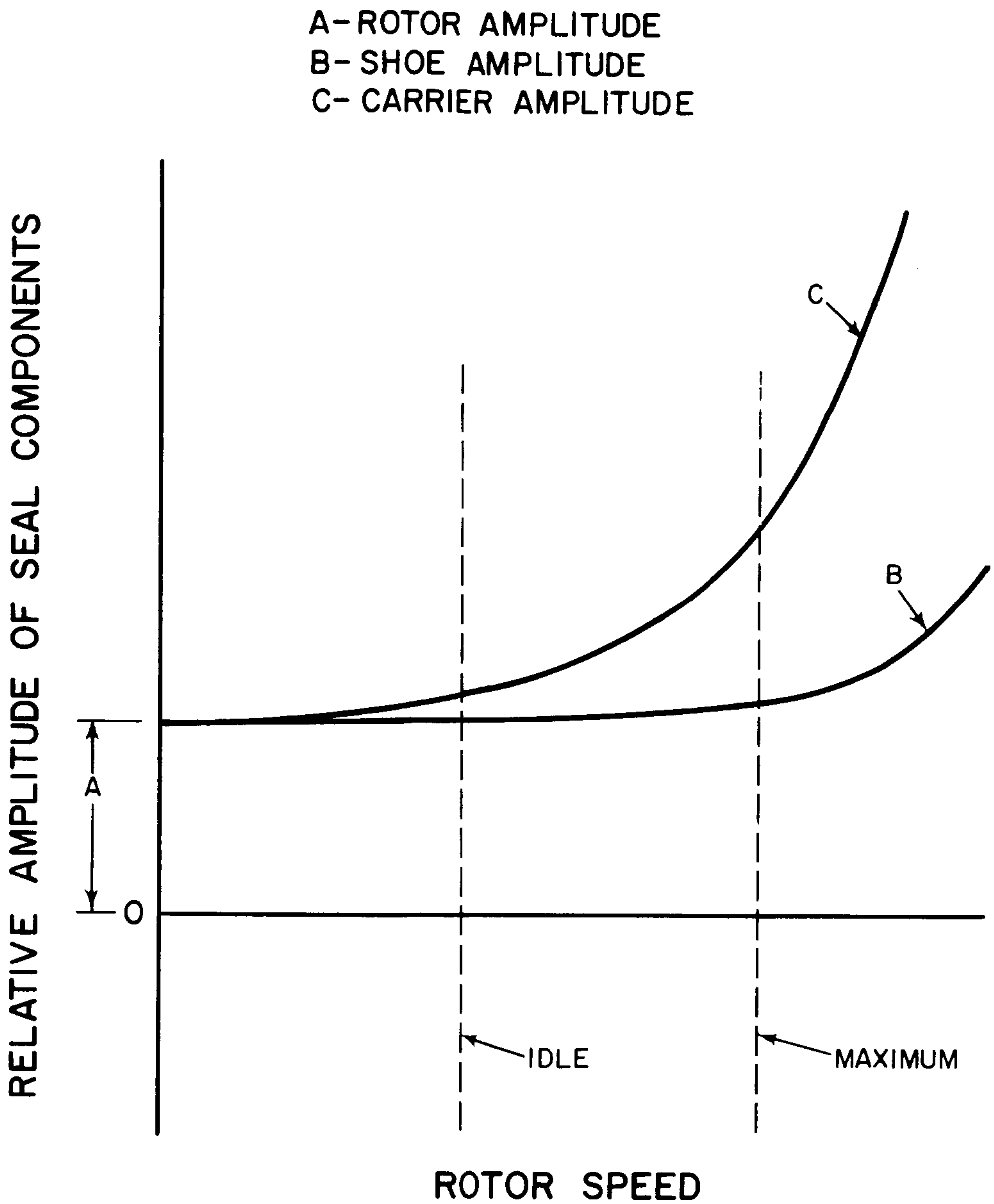


Figure 11 Variation of One-Side Floated Shoe Seal Vibration Amplitude with Rotor Speed for a Light Carrier and Stiff Wave Spring

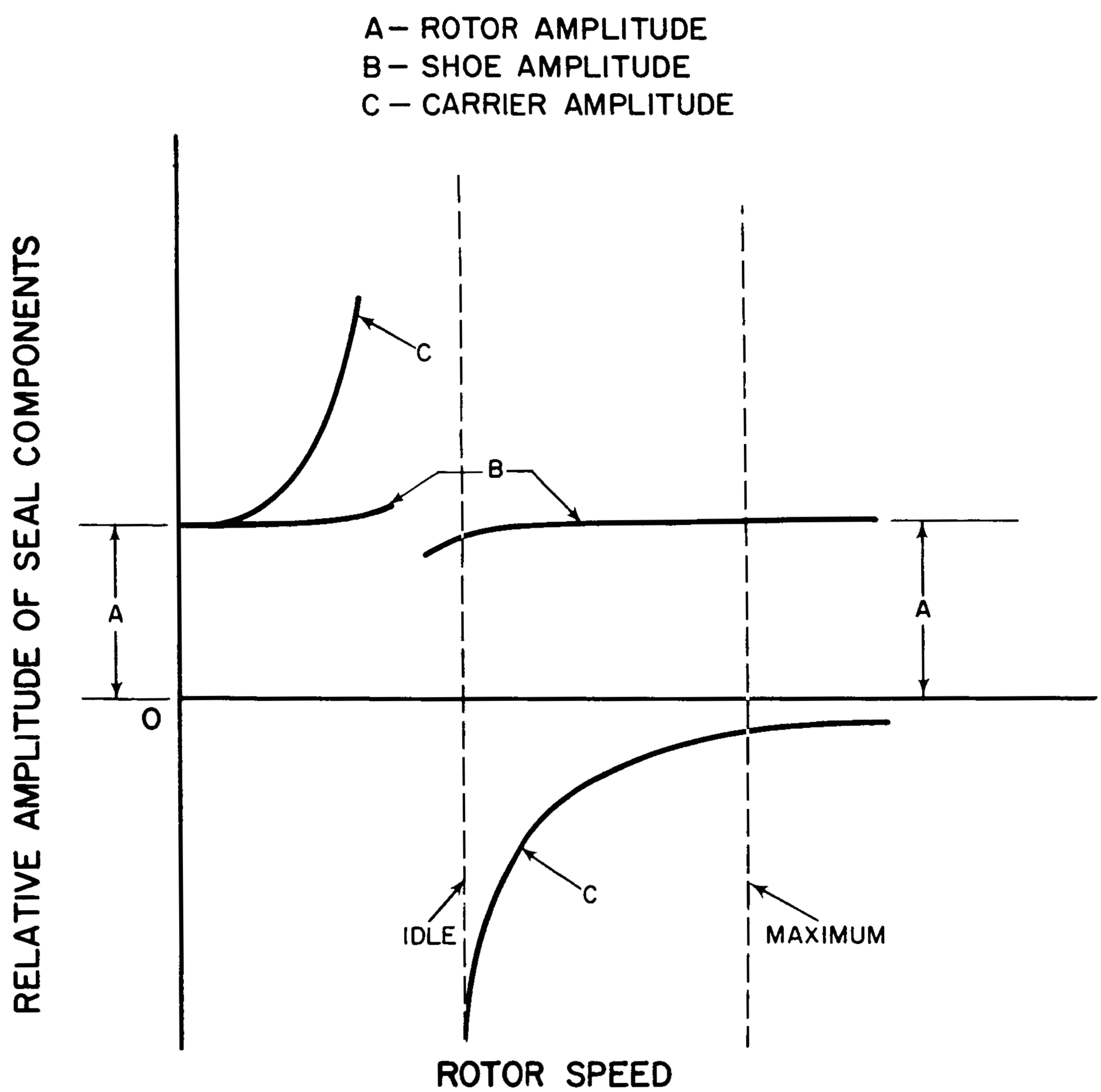


Figure 12 Variation of One-Side Floated Shoe Seal Vibration Amplitude with Rotor Speed for a Heavy Carrier and Stiff Wave Spring

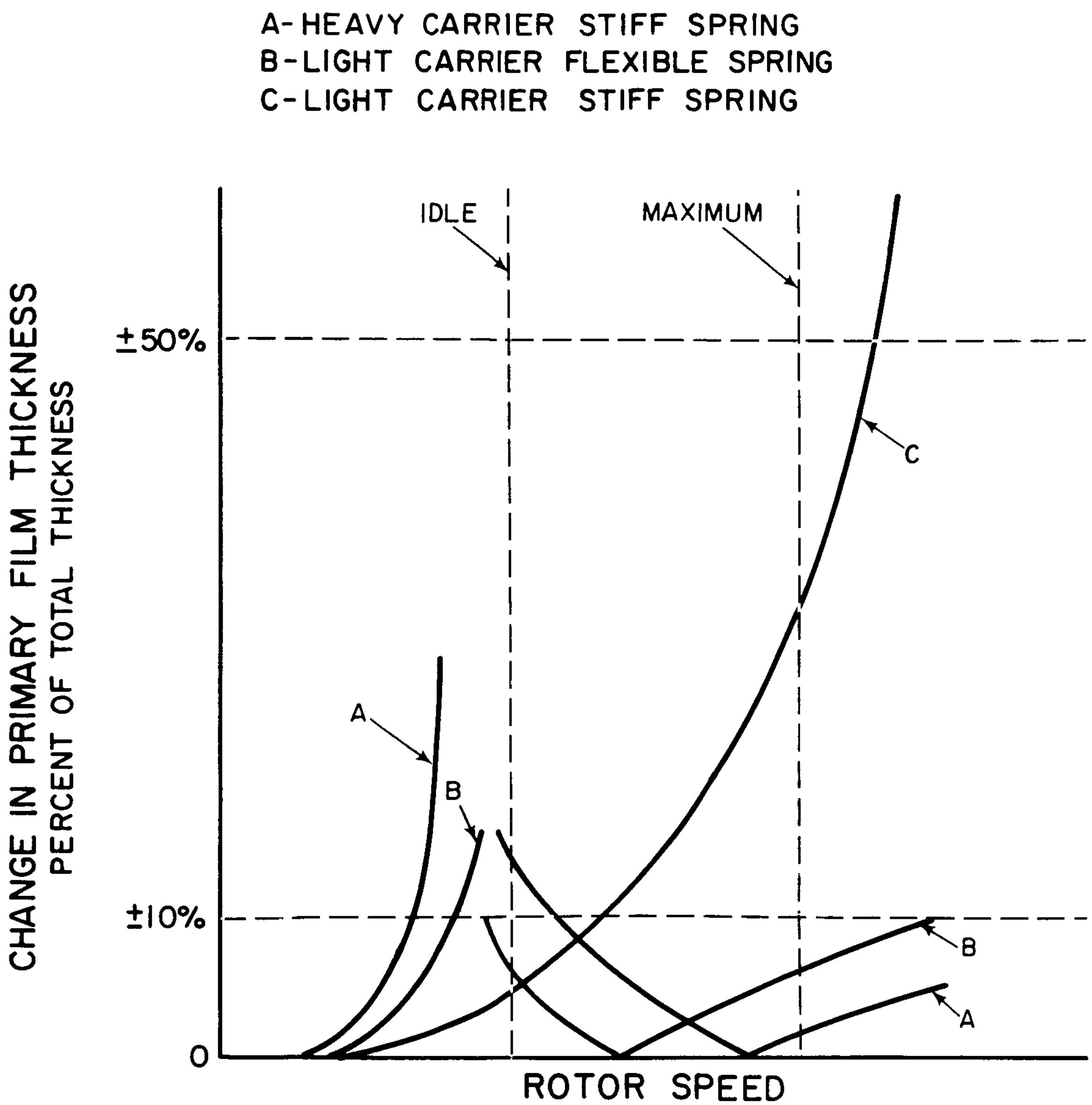


Figure 13 Effect of Carrier Weight and Shoe-to-Carrier Spring Rate on Primary Film Thickness at Maximum Rotor Runout

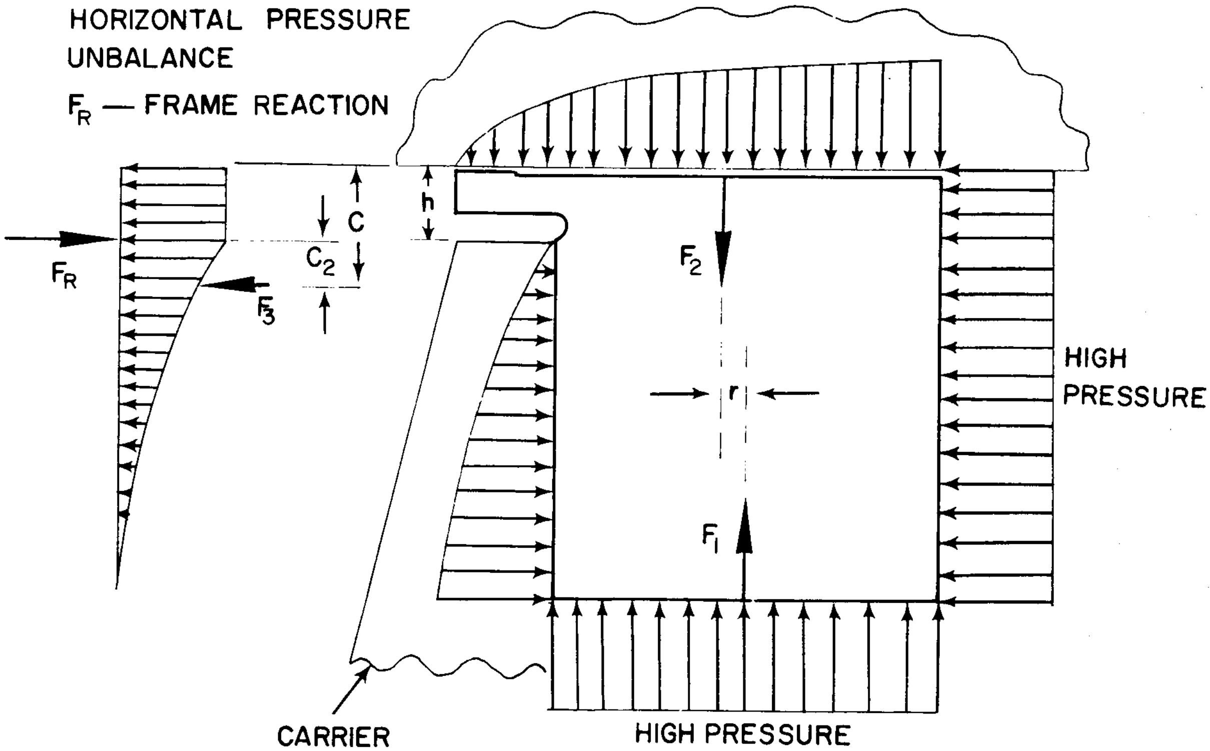
A study of seal tolerances and clearances has shown that the concentricity between primary seal diameter and support secondary seal diameter is critical. A lack of concentricity of the carrier with respect to the carrier support will result in a local change in force balancing areas on the carrier. A 0.005 inch radial displacement of the carrier creates a tipping force or moment on the carrier, which is resisted by the carrier-shoe spring. The result is a carrier-shoe spring deflection which varies by approximately ± 0.030 inches around the circumference at sea level take-off conditions. Further allowances for acceleration loading and carrier travel requirements place the total demand for shoe-to-carrier displacement at approximately ± 0.040 inches from the nominal position. When the shoe-to-carrier piston ring is axially fixed with respect to the carrier, and the shoe-to-carrier displacement exceeds ± 0.020 inches, serious deviations of the secondary seal film thickness occur.

In order to accommodate 0.080 inches of travel of the shoe on the carrier, the shoe-to-carrier piston ring will be fixed with respect to the shoe. This change can be facilitated by inverting the shoe so that the shoe-to-carrier secondary seal surface is transferred from the inner surface of the shoe assembly to the outer surface. Inverting the shoe eliminates the need for an undercut at the inner edge of the rotor surface, resulting in a reduction in rotor weight.

F. PISTON RING GEOMETRY

The requirement that frictional drag on all secondary seal surfaces be minimized has led to the design of hydrostatically floated piston rings for the shoe-to-carrier and carrier-to-frame applications. Design of the piston rings has been further complicated by the necessity for thin cross-sections and large diameters.

Figure 14 shows the forces acting on the carrier-to-frame piston ring. The vertical forces are balanced with the ring floating on a thin gas film which derives its load-carrying capacity from a hydrostatic step seal configuration. The horizontal pressure forces are supported by contact with the carrier. The ring is moment-balanced for proper seating of the ring in the carrier groove. When relative radial movement occurs between the carrier and the frame, the radial gas loads on the ring will overcome the friction force between the ring and the carrier, providing a nearly constant air film thickness between the piston ring and the frame. Design of the shoe-to-carrier piston ring is based on the same approach.



FORCE BALANCE
$F_1 = F_2 = T_1$ AND $F_3 = F_R = T_2$

MOMENT BALANCE
$T_1 \times r = T_2 \times C_2$

WHERE $C_2 = C - h$

h= CLEARANCE BETWEEN CARRIER AND FRAME
C=CENTROID OF PRESSURE AREA

Figure 14 Force and Moment Balance of Piston Ring

G. SPRING AND TORQUE PIN DESIGN

Two mechanical spring designs are used in the seal assembly. The first, a wave spring having an axial stiffness of 44 pounds per inch of circumference (3750 pounds per inch total), positions the shoes with respect to the shoe carrier. The second is a coil spring of relatively low stiffness: 24 of these are used to seat the carrier and shoes against the rotor. Torque constraint of the carrier is provided by four torque pins attached to the carrier support.

The wave spring has been designed as a flat strip of metal trapped between the shoe and the carrier. Tabs on the spring are fitted into notches in parts attached to the carrier and shoe assembly. The notched parts serve as load-bearing points for the axial spring load. The relationship between axial load, stress, and deflection is shown in Figure 15.

The 24 coil springs are spaced evenly around the circumference to provide the axial force necessary to seat the carrier and shoes against the rotor without causing excessive friction and wear during starting. These springs must allow 0.4 inches of travel of the carrier with respect to the frame, in order to accommodate changes in rotor position during engine operation. A low spring rate (36 pounds per inch total for 24 springs) was chosen to reduce the effect of rotor position on seal operating characteristics. With both ends fixed, the lowest resonant frequency of the coil spring is 9380 cycles per minute, which is above the maximum rotor speed. For this reason, fatigue failure of the coil spring is not likely to occur. Figure 16 describes the operating characteristics of the spring at room temperature and at temperatures corresponding to take-off and cruise conditions.

Four torque pins attached to the carrier support are used to provide torque constraint on the carrier. Each torque pin is machined with two eccentric centerlines so that rotation of the pin about one centerline changes the position of the other centerline with respect to the support. This device permits the carrier position to be adjusted to the high degree of concentricity with the support that is required when the carrier-to-support piston ring is made part of the carrier assembly.

The carrier end of the torque pin is fitted with a sleeve which has parallel flat external surfaces. The sleeve fits into radial slots in a flange on the carrier. The slots provide freedom for the carrier to expand or contract with respect to the frame, in response to thermal gradients and pressure loads.

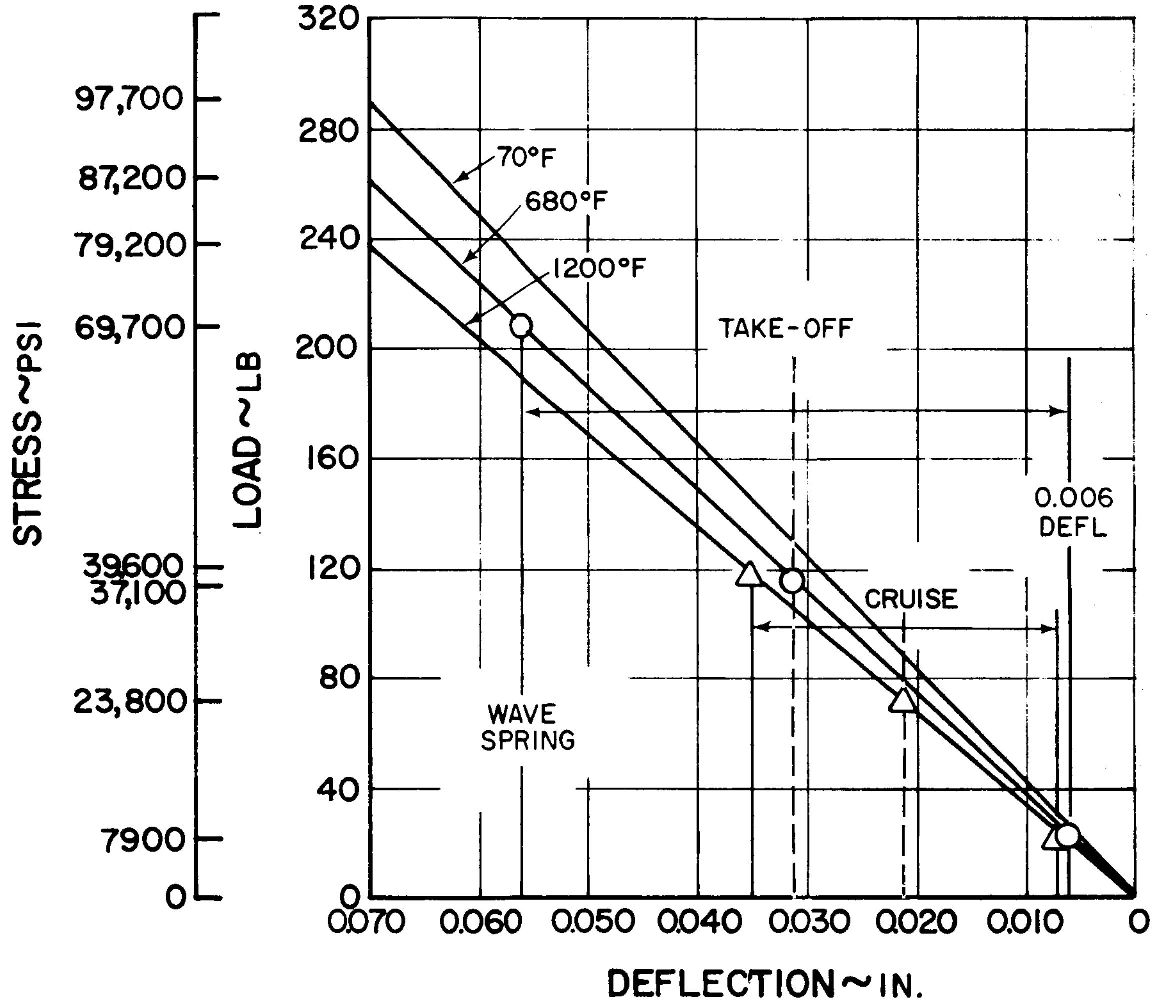


Figure 15 Deflection of Carrier Wave Spring.
Material: Waspalloy (AMS 5544)

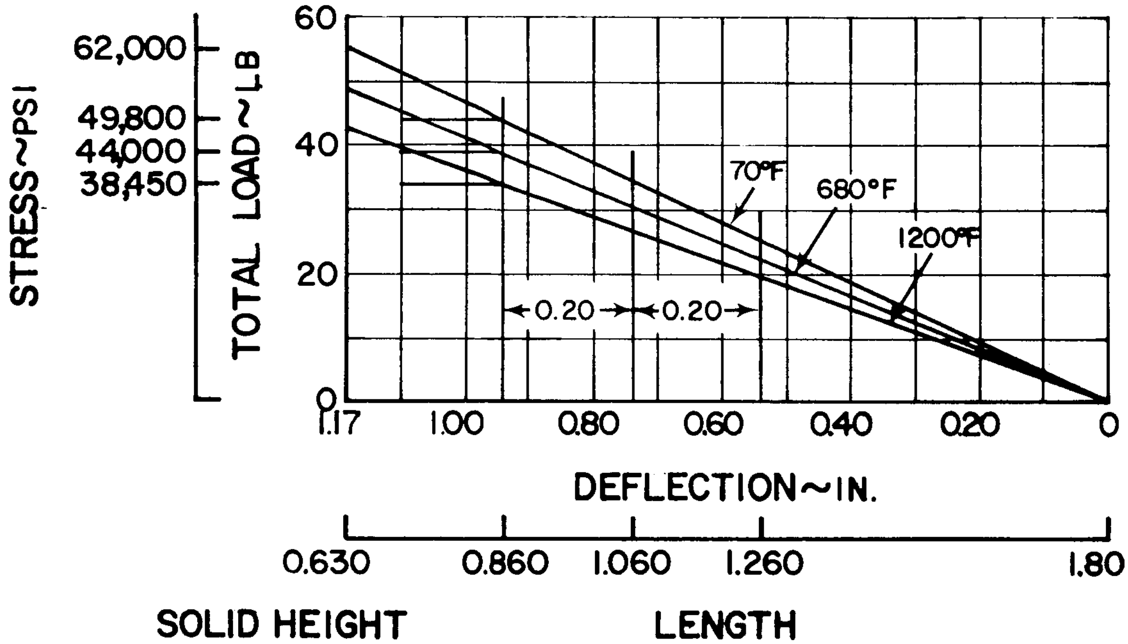


Figure 16 Deflection of Carrier Coil Spring.
Material: Inconel X-750 (AMS 5699)

H. CARRIER SUPPORT DESIGN

The contractor has completed the design of the carrier support. It was designed with the same critical distortion limits that were applied to the carrier itself. The support will have no distortions exceeding the 0.001 radian slope limit, which is considered to be reasonable for this hydrostatically floated seal device. The carrier support is that part of the seal assembly which is attached directly to the engine frame, and serves to integrate the carrier springs and torque pins and the secondary seal into a single major subassembly.

The distortion problem is eased slightly because thermal distortion of the carrier support produces a deflection which is in the same direction as the deflection induced by thermal distortion of the carrier. The deflection and slope are critical because the piston ring secondary seal between the carrier and support is sensitive to the relative slope between the two parts.

The deflections induced by pressure loading have been particularly difficult to control because of a large variation in the axial position of the carrier which alters the pressure loading on the carrier support. A further complication is the requirement that the secondary sealing surface of the carrier support must be isolated from slope and radial displacement at the point of attachment to the engine frame. The contractor has met this requirement by placing the secondary sealing surface on the inside of a relatively thick cylinder (0.30 inches thick) and joining that cylinder to a long thin cylinder (0.06 inches thick) through a flange on the outer surface of the thick cylinder. The flange is located approximately at the midpoint of the secondary seal axial travel.

I. ROTOR DESIGN

Design of the seal rotor has been completed. It has been governed by the same distortion limits which were applied to the stationary seal parts. The distortions induced by pressure and centrifugal loading were corrected separately, but the rotor was designed so that the feature which reduces centrifugal distortion is also effective in controlling the elastic distortion due to thermal gradients.

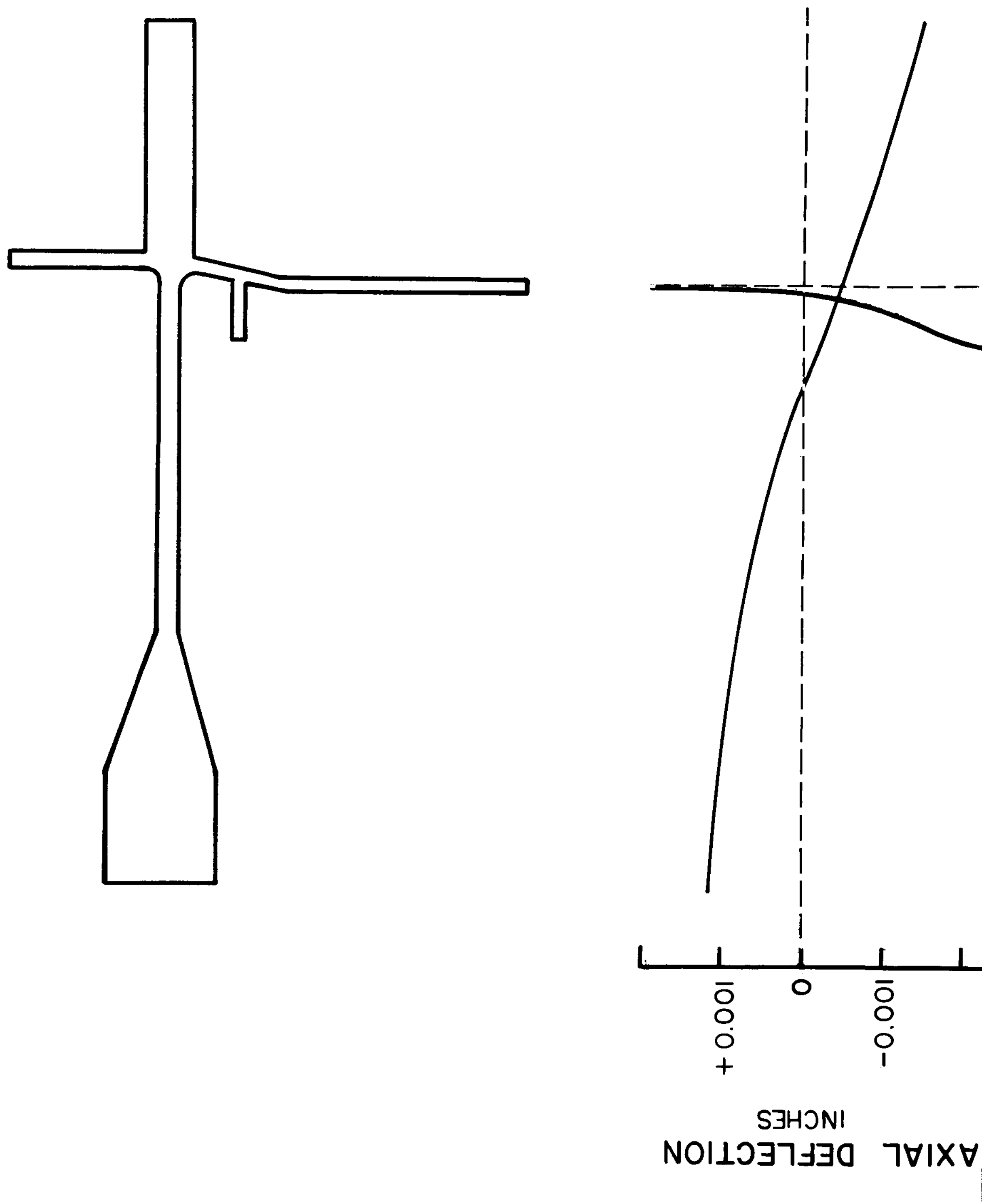
It was recognized in the conceptual design effort that the design and the testing of the seal rotor would be greatly facilitated by isolating the rotor from the bladed compressor discs. A thin cylinder joining the seal rotor to the compressor rotor makes the seal insensitive to axisymmetric distortion of the compressor rotor structure. The thin cylinder must be matched to the seal rotor configuration so that pressure loads will not cause the face of the seal rotor to deviate significantly from a flat surface. When the thin cylinder is subjected to radial pressure loading, it experiences negative radial deflections, which are transmitted to the seal rotor as shear and moment loads. The pressure-loaded regions of the rotor disc must then be positioned so that a moment is created to balance the moment produced by the thin cylinder. Figure 17 describes the seal rotor deflections induced by pressure.

To counteract centrifugally induced deflections of the seal rotor, a second thin cylinder has been located on the side of the rotor opposite to the first cylinder. The reasoning which led to this design feature is as follows:

- Centrifugal loading on the thin cylinder joining the seal rotor and the compressor causes the cylinder to expand radially.
- Radial expansion of the cylinder applies shear and moment loads to the seal rotor.
- A second cylinder, opposite to the first, will apply opposite shear and moment loads to the seal rotor, balancing the loads applied by the first cylinder.

The second cylinder is less than half as long as the first, but has approximately the same thickness and radius. Figure 18 describes the centrifugally induced deflections of the seal rotor.

The thermal growths of the seal rotor and the thin cylinders follows the pattern of the centrifugal growth. The growth of the seal rotor is constrained by the cooler material located near its inner surface. The two cylinders apply equal and opposite moments to the rotor, since their temperatures are approximately equal. Figure 19 describes the thermally induced deflection of the seal rotor.


+0.001
0
-0.001
AXIAL DEFLECTION
INCHES

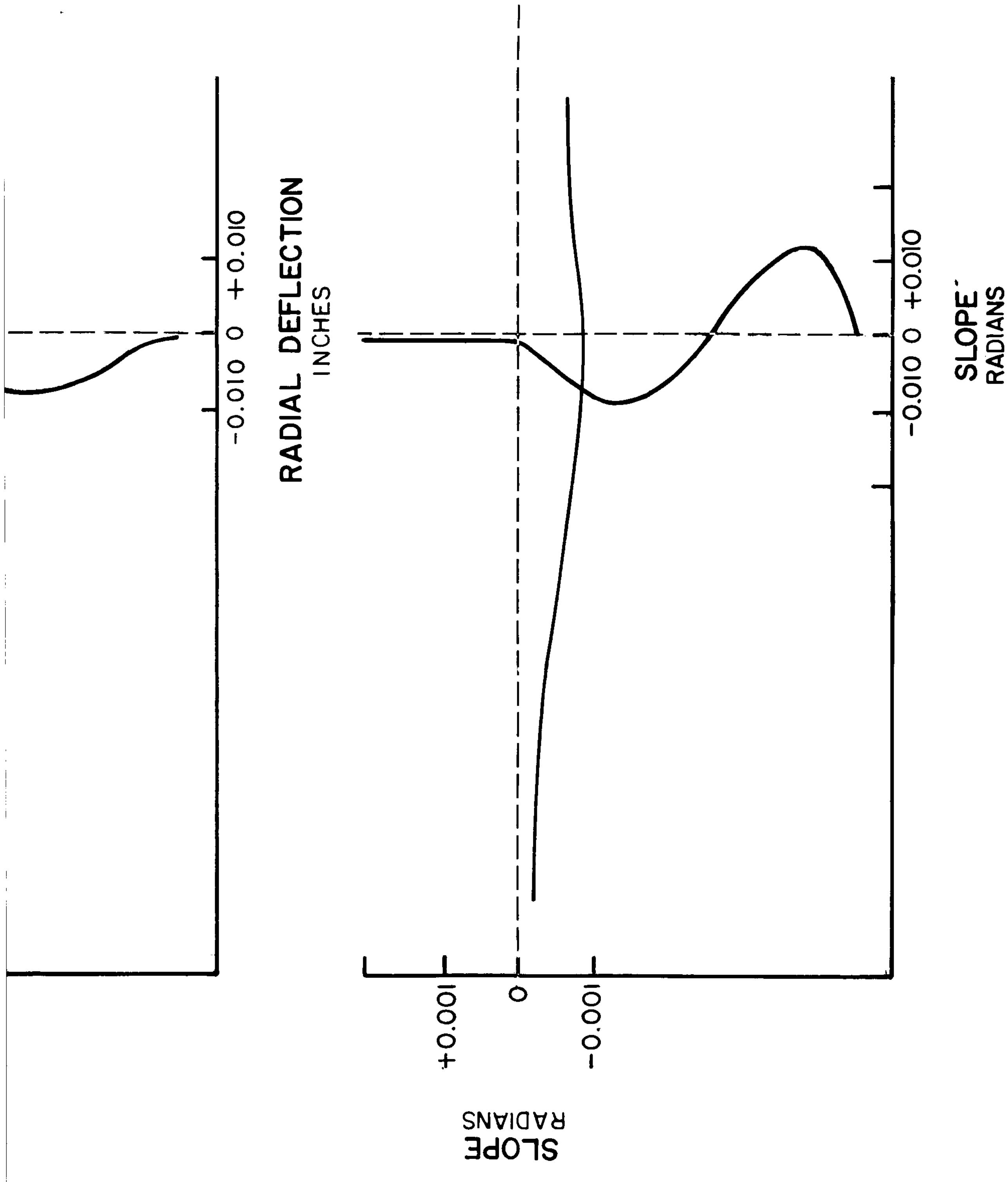
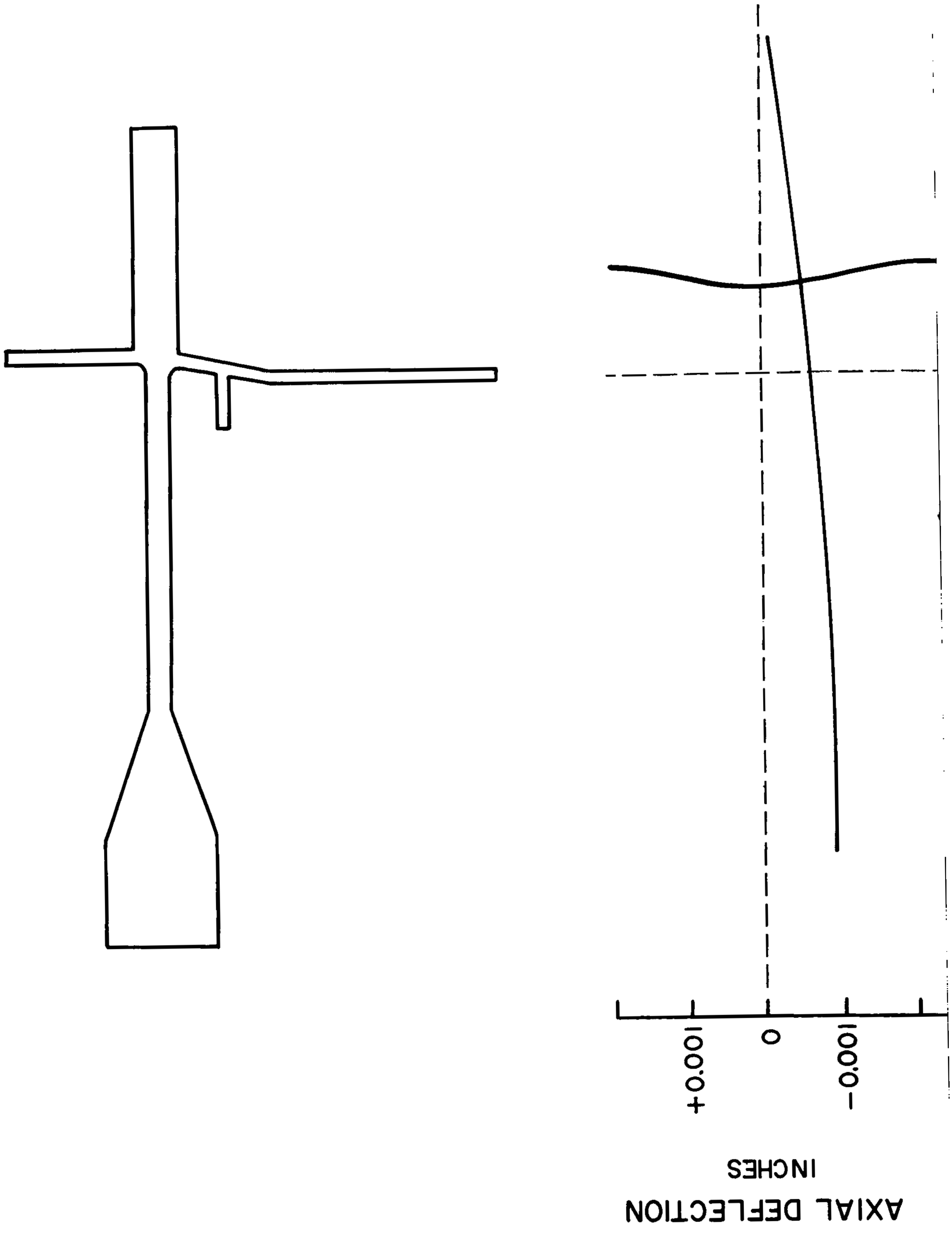


Figure 17 End Seal Runner Deflections under Pressure Loading


+0.001
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-0.001
AXIAL DEFLECTION
INCHES

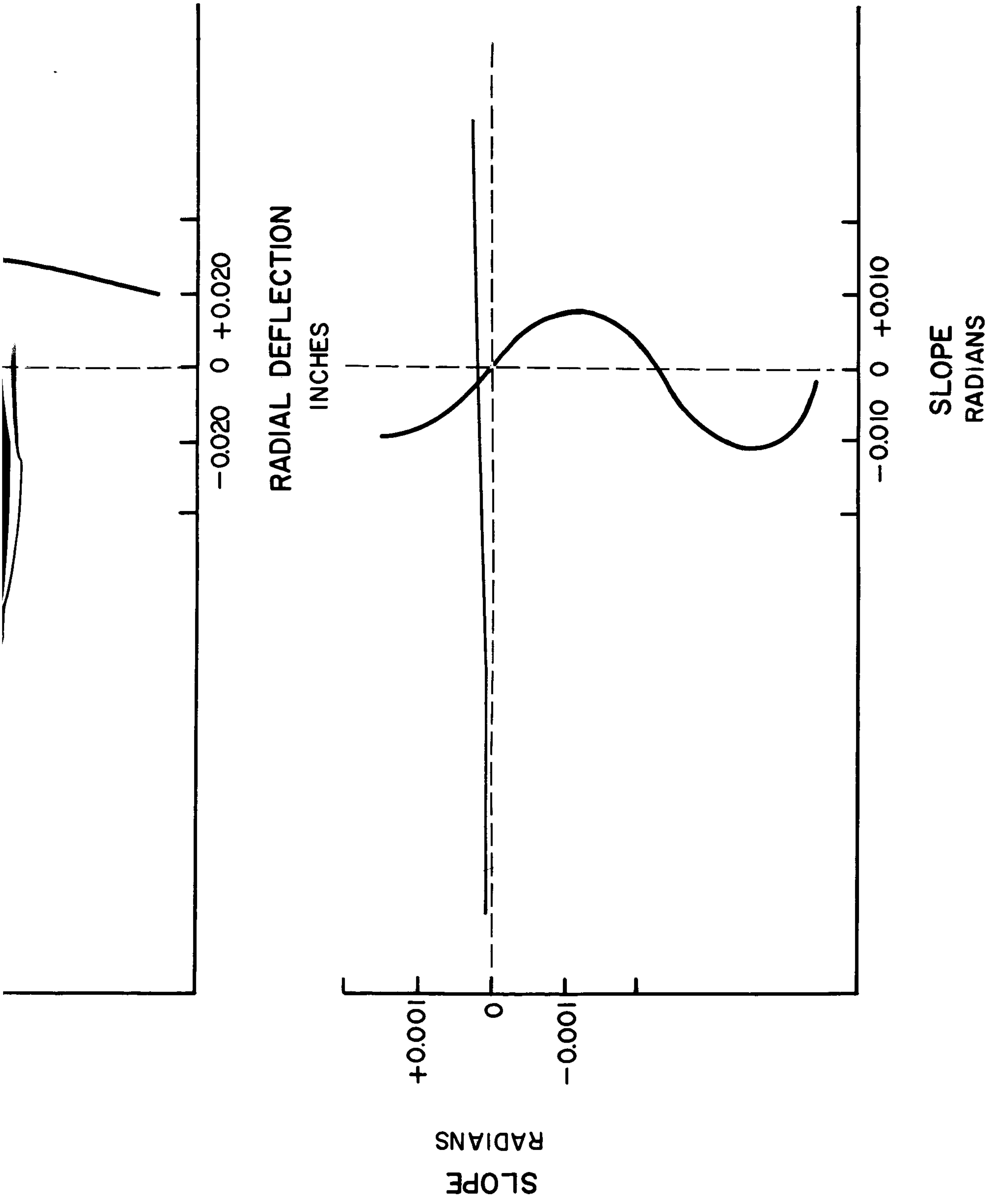
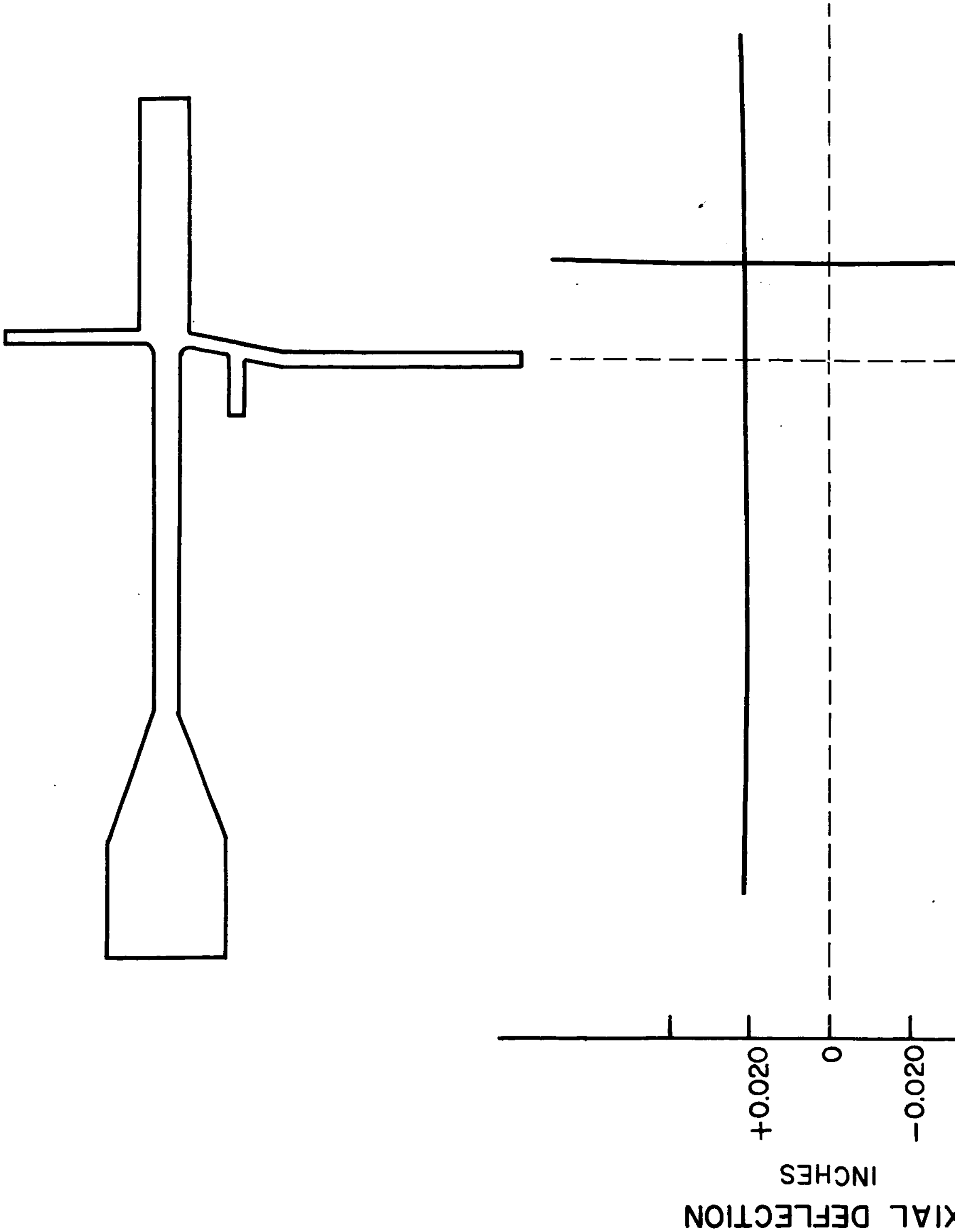


Figure 18 End Seal Runner Deflections under Centrifugal Loading


+0.020
0
−0.020
INCHES
DEFLECTION

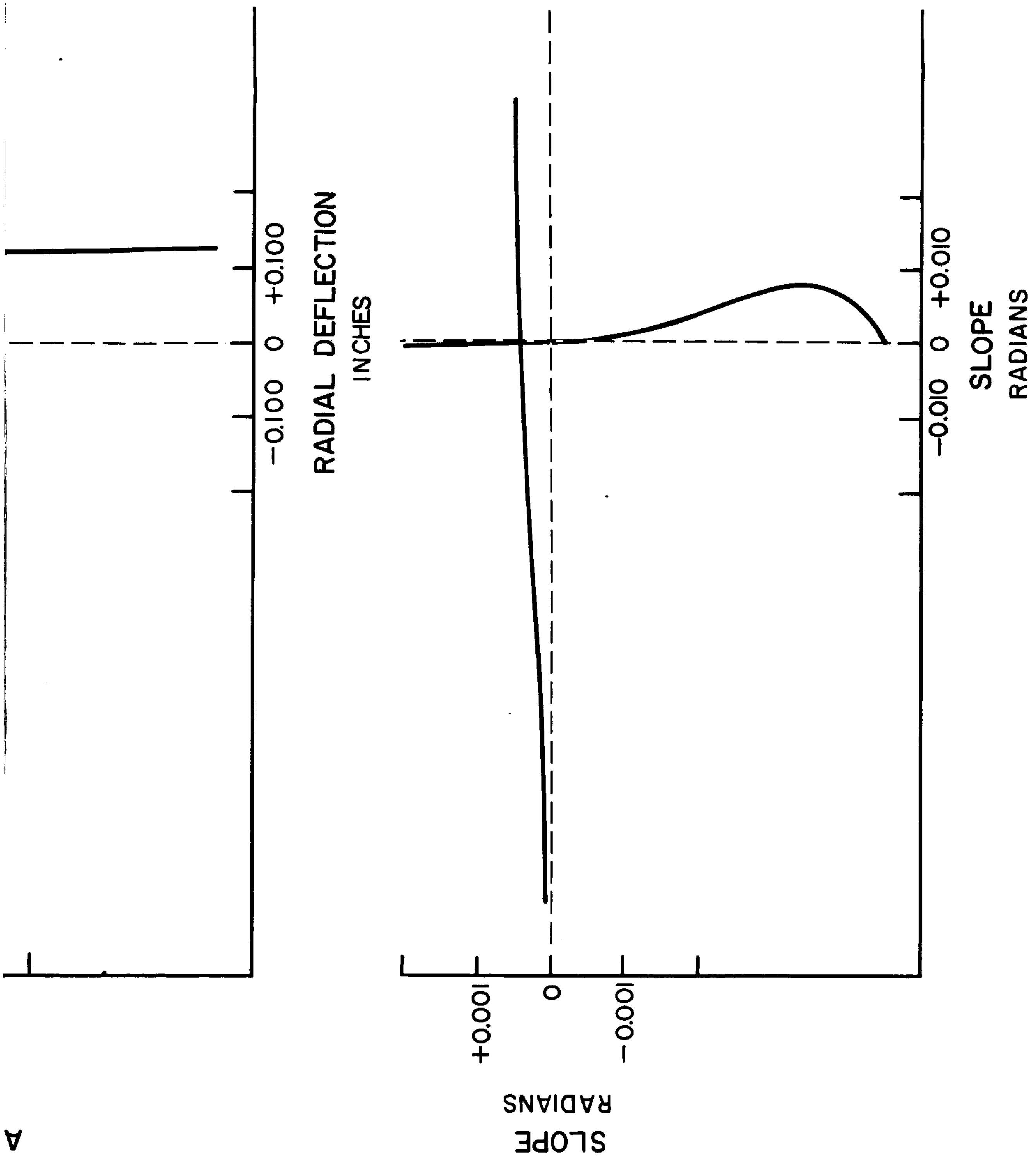


Figure 19 End Seal Runner Thermal Growth

J. DYNAMIC ANALYSIS

The contractor has succeeded in reducing the masses of the seal shoes and the carrier, so that the first resonant frequency occurs at 2700 revolutions per minute, well below engine idle speed. The principles governing the dynamic response of the seal system were outlined in the second Semiannual Report (PWA-2875), on pages 64 through 69. The final design has altered the mass and stiffness values which were assumed in that report. Primary film stiffness at the design point operating condition has increased, and the wave spring and coil spring stiffnesses have decreased, as described in sections II. C. and II. G. of this report. Because of the changes to the design, the dynamic response of the shoes to the rotor has improved significantly. Figure 20 describes the amplitude variation with speed for the shoes and the carrier, assuming zero damping.

The current approach to carrier damping implies that there will be a limited amount of shoe contact with the rotor at 2700 revolutions per minute. The contractor has taken this approach because any increase in friction between the carrier and carrier support can seriously overload or underload the wave spring when the axial position of the carrier on the support changes by more than 0.001 inches. This approach is not considered to be dangerous because the rotor speed will not dwell at the resonant frequency for any appreciable length of time. The windmilling speed of the engine during an in-flight shutdown will have to be determined experimentally for each specific engine design so that final adjustments to the resonant frequency can be made as required to avoid prolonged excitation.

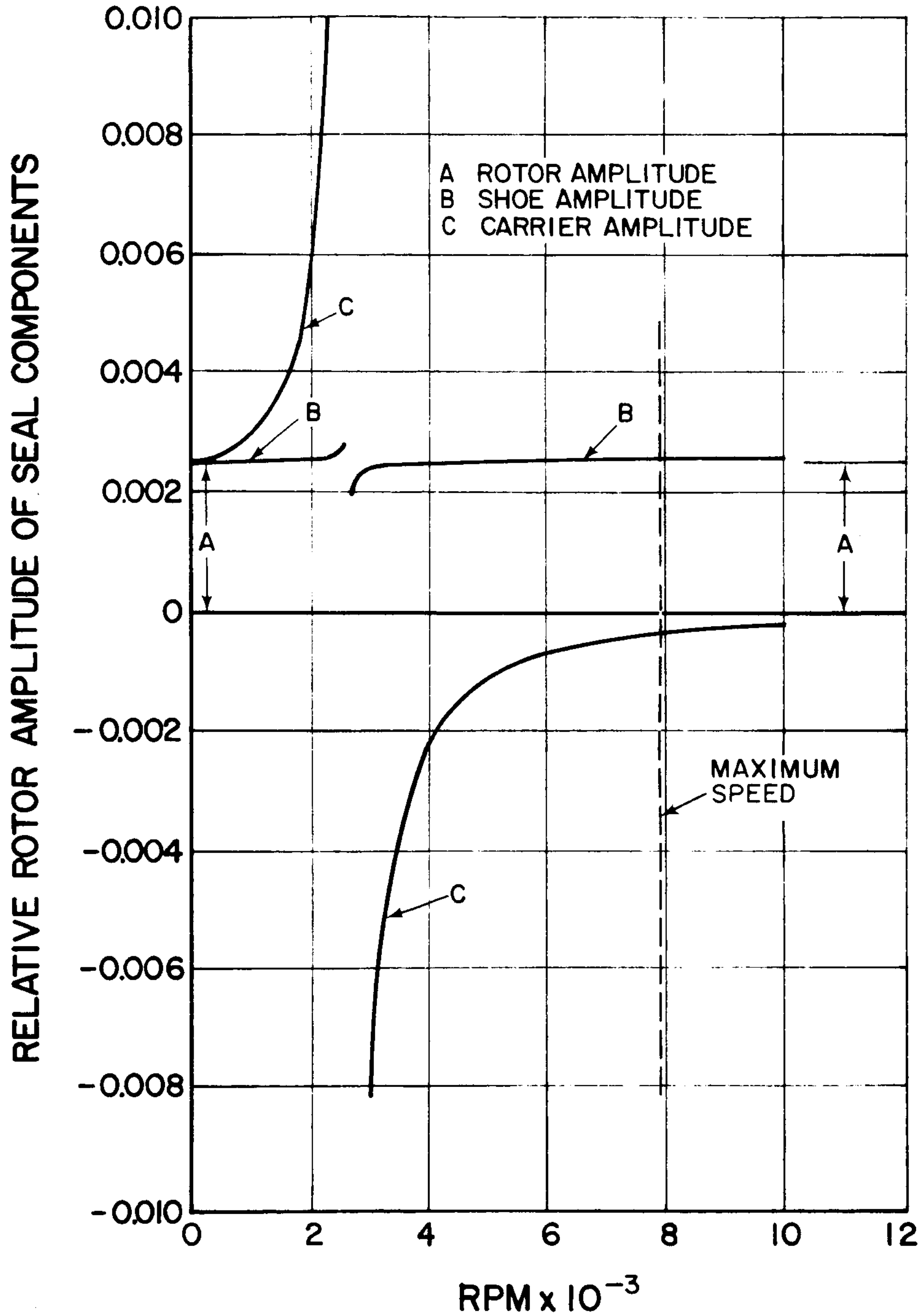
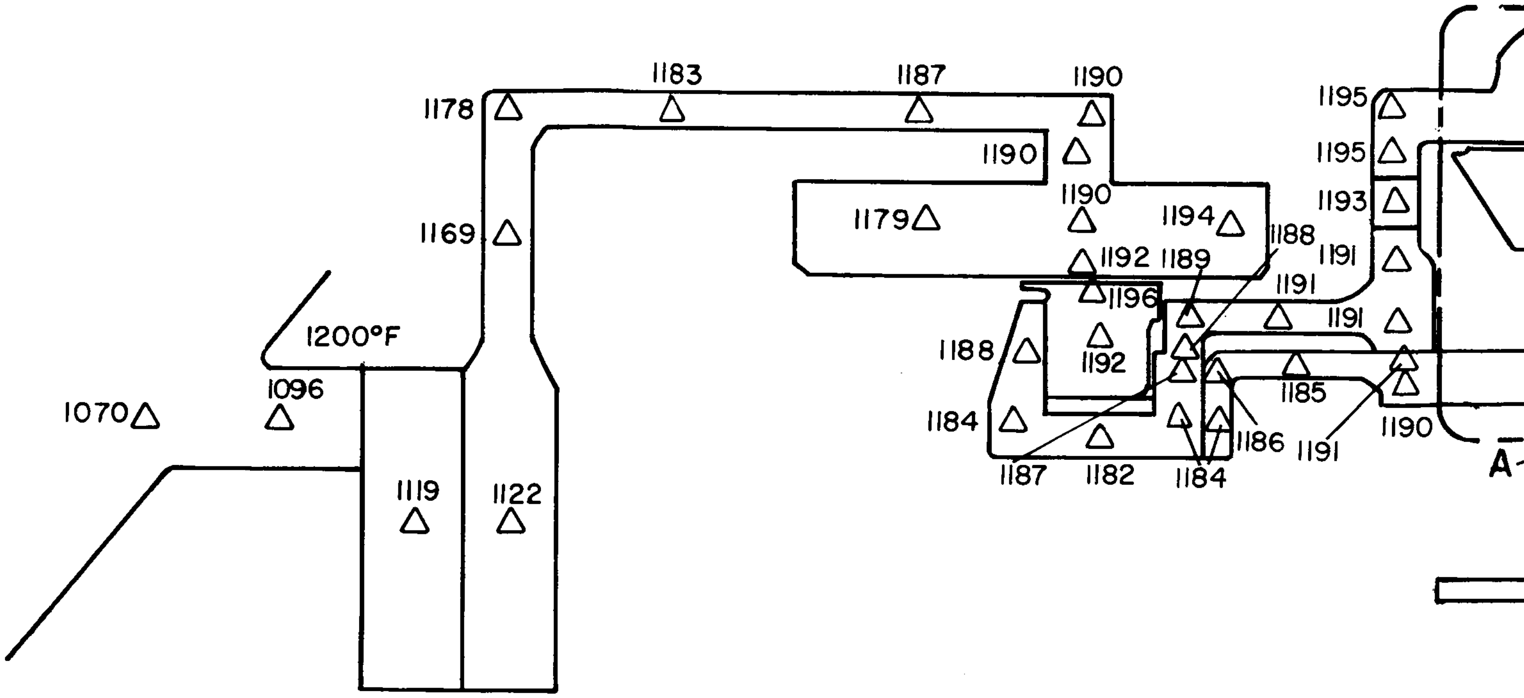


Figure 20 Dynamic Response of the Shoe and Carrier, Final Configuration

K. THERMAL ANALYSIS

Analytical studies were conducted during this report period to define the thermal map for the redesigned seal. These studies have established the basis for the thermal distortion studies described in earlier sections of this report. The thermal map is shown in Figure 21.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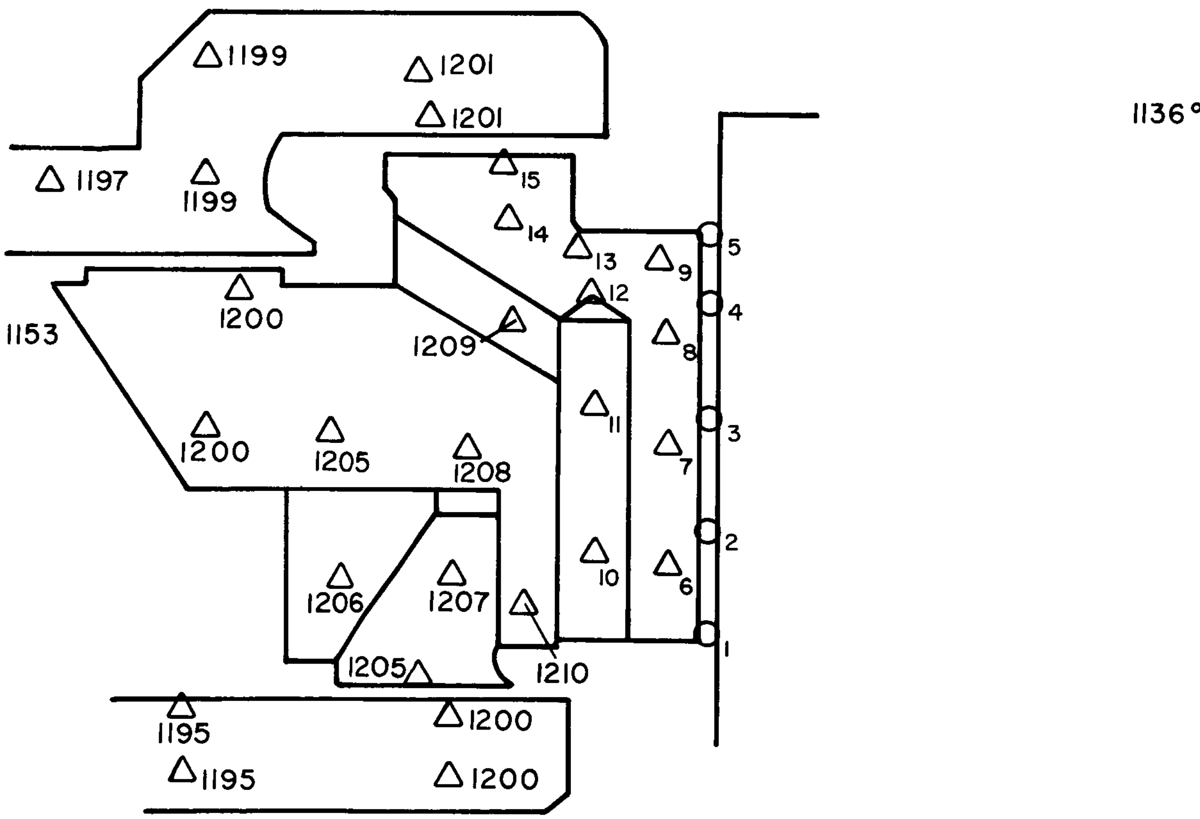
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1200

VIEW A

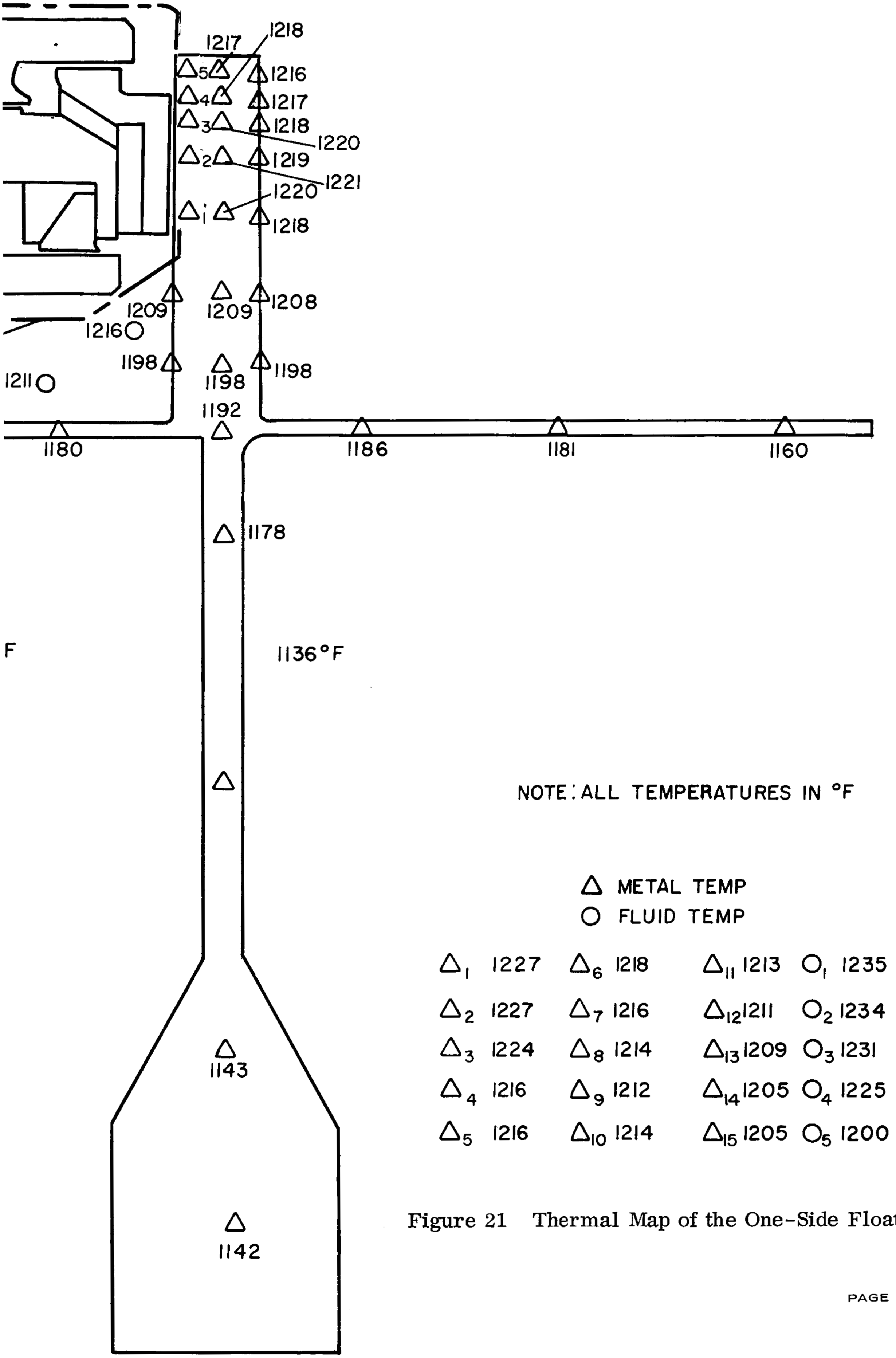


Figure 21 Thermal Map of the One-Side Floated Shoe Seal

L. COMPRESSOR SEAL TEST RIG

1. FRONT BEARING COMPARTMENT

The front bearing (roller) compartment layout has been completed. The oil scavenge system has been designed so that the oil lines will not interfere with the removal of the outer case for seal measurements. A heat shield has been provided to protect the carbon seal from the high temperatures anticipated in the test compartment. A 24 channel slip ring assembly adjacent to the front bearing compartment will be used to monitor disk, hub, and seal plate temperatures. A deflection adapter for the cantilever section of the slip ring has been incorporated into the design, with provisions made for alignment.

2. THRUST BEARING COMPARTMENT

The design of the seal rig thrust bearing compartment has been completed. It is an adaptation of an existing duplex bearing and seal configuration which has been very reliable in experimental testing and field service. The bearing compartment and gearbox have been designed with a common oil supply, eliminating the need for a carbon face seal to separate the two areas. The common oil system will include both scavenge and breather systems. The face seal will be replaced by an O-ring.

The design also incorporates a double-acting piston arrangement on the rear set of thrust bearings to compensate for the high thrust loads from unbalanced rotor forces. This device uses air from a rig compartment, eliminating any need for a separate air supply and regulator. Because of high operating loads, bearing temperatures will be monitored through thermocouples at the outer races.

3. ROTOR

The rotor design has been completed, with the exception of the end and interstage seal runners. The rotor tie bolt analysis has been completed, accounting for thermals, thrust loads, Poisson's effect, and the elastic growth of the members. Provisions have been made for rotor assembly balancing. Each rotor part will undergo an individual detail balance. The disks will be balanced by end drilling at the outer circumference, while weights will be added to hubs and spacers for balance purposes. The entire assembly will then be balanced in two planes by weight addition and removal.

Procurement of the rig rotor forgings has been initiated with the approval of the NASA contracting officer. Calculations have been made for spin test disk shapes of the rig rotor forgings. Anticipated stresses and growths have been calculated, and will be compared with results obtained from the spin tests. Existing fixtures from previous high pressure compressor spin tests will be used wherever possible.

4. OUTER CASE

Design of the rig outer case is in progress, with the rig operating pressures, temperatures, deflections, and loads being taken into consideration. Overshoot pressure limits have been established so that provisions can be made for a relief valve system. To provide for the required 0.25 inches of axial travel of the case with respect to the rotor, a ball bearing and piston ring arrangement has been designed, using a one-row ring of balls. A previous three-row design was discarded because of the difficulty and expense of providing an adequate bearing cage.

The axial travel actuator design has been completed. A mounting ring is attached to the outer race of a bearing supported outside the rig case. Six links connect the ring to the case. As the ring is rotated, the case is moved in an axial direction, rotation being prevented by a device similar to a spline.

5. TARGET DATE FOR COMPLETION OF DESIGN

It is anticipated that the test rig design will be completed by 31 January 1967, after which NASA approval will be requested and detail drawings will be started.

III. TASK III
COMPRESSOR STATOR PIVOT BUSHING AND SEAL CONCEPT FEASIBILITY ANALYSIS

A feasibility analysis program was conducted on stator vane pivot bushing and seal concepts for application in compressors for advanced air-breathing propulsion systems. The first phase of this program consisted of a preliminary analysis and a screening of various seal concepts prior to the selection of concepts for the detailed feasibility analysis. The analytical effort included a comparison of the selected concepts to current practice and all calculations, analyses, and drawings necessary to establish feasibility of these selected concepts. This analytical program was subcontracted to Mechanical Technology Inc. of Latham, New York and was monitored by Pratt & Whitney Aircraft as required under the terms of the NASA contract.

Work under Task III has been completed. Pratt & Whitney Aircraft submitted the latest feasibility designs of the single bellows and spherical seat vane pivot seals to NASA on 19 May 1966 requesting approval to start final design of these seals under Task IV. An effort was made to simplify the seal designs within practical limits without making major changes in the basic seal concepts shown on the MTI drawings. Approval was granted in a letter from NASA dated 31 May 1966. A more detailed summary of the work performed under Task III is contained in the first two semiannual reports (PWA-2752 and PWA-2875).

IV. TASK IV
PIVOT BUSHING AND SEAL EXPERIMENTAL EVALUATION

A. INTRODUCTION

This phase of the program provides for final design and procurement of bushings and seals, design and fabrication of a test rig, and experimental evaluation of bushing and seal assemblies.

The final design of the two concepts selected for experimental evaluation includes all calculations, material determinations, analyses, and drawings necessary for pivot bushing and seal optimization, procurement, and experimental evaluation.

A single-vane test rig has been designed and will be fabricated to evaluate the two selected pivot bushing and seal designs under simulated operating conditions for the last compressor stage. The actuating mechanism and vane are applicable to current advanced engine practice.

The pivot bushing and seal assemblies will be calibrated in incremental steps over the full pressure and temperature range, with a maximum pressure of 135 pounds per square inch and a maximum temperature of 1200 degrees Fahrenheit.

The seals will be subjected to a cyclic endurance run of at least 40 hours duration following a test program which provides for simulation of take-off (20 hours) and cruise (20 hours) conditions typical of advanced engine designs through duplication of:

- Compressor stage air temperatures
- Supporting structure geometry
- Supporting structure temperatures
- Pivot movements as required for the vanes
- Pivot loading (mechanical loading to simulate air loading is acceptable)
- Compressor stage pressure drop

The pivot movement will be a minimum of 13 degrees at 10 cycles per minute. The pivot loading will include a vibratory load at a convenient frequency superimposed on the steady load and equal to approximately ±15 percent of the steady load.

In a letter dated 9 September 1966, NASA approved the final designs for the single bellows vane pivot seal, the spherical seat vane pivot seal, and the test rig. On 5 October 1966, NASA granted approval to incur hardware expenses for the seals and the rig. Hardware procurement is now in progress. The contractor expects to start testing by 1 April 1967.

B. SEAL AND TEST RIG DESCRIPTIONS

Final design layouts and detail drawings were completed for the single bellows and spherical seat vane pivot seals and the test rig in which the seals will undergo experimental evaluation. The seal and rig designs are shown in Figures 22, 23, and 24. All instrumentation, materials, and coatings are indicated in the designs. In some cases, alternate materials are provided for relative evaluation during the test program. The test rig and seal designs are described below.

Electrical heating is provided in the test rig housing to maintain the required air temperature surrounding the simulated vane and entering the seal. No attempt is being made to pre-heat the air entering the test chamber, since the leakage rates are expected to be small. Since the bellows seal of the vane loading mechanism would be overstressed at high pressure differentials, air of equal pressure is admitted around it. This air will leak to atmosphere around the shaft, providing cooling. The flow will be appreciable, but it is not in the test seal leakage measurement circuit.

The bending moment, both steady and vibratory, is applied by a cam mechanism. Vibratory deflection is built into two cams, one for cruise, the other for take-off. Steady bending load adjustment is obtained by movement of the cam shaft centerline with respect to the seal or vane centerline. Loading will be checked by means of a strain gage on the thin wall tubular section of the push rod. It is expected that both steady and vibratory loading can be measured with this calibrated system. Temperature will be monitored in the vicinity of the strain gage and a correction applied to the readings if necessary.

Vane actuation is provided through a lever system. Measurement of vane actuation torque is accomplished by means of a strain gage on the flat portion of the lever. The effect of deflection in this section on pivot travel has been compensated using calculated pivot torque. As a result, ±0.030 inch overtravel of the lever tip is designed into the system so that the required 13 degrees of pivot motion is achieved.

Seal leakage will be measured externally as the flow required to maintain constant pressure in the test cavity. Pressure taps are provided to measure cavity pressure. Thermocouples are provided as indicated to measure air, bushing, seal and supporting structure temperatures.

The following materials and coatings are provided at various locations on the test parts to minimize galling and wear. Where several coatings are listed, alternates are being provided for evaluation in the test program.

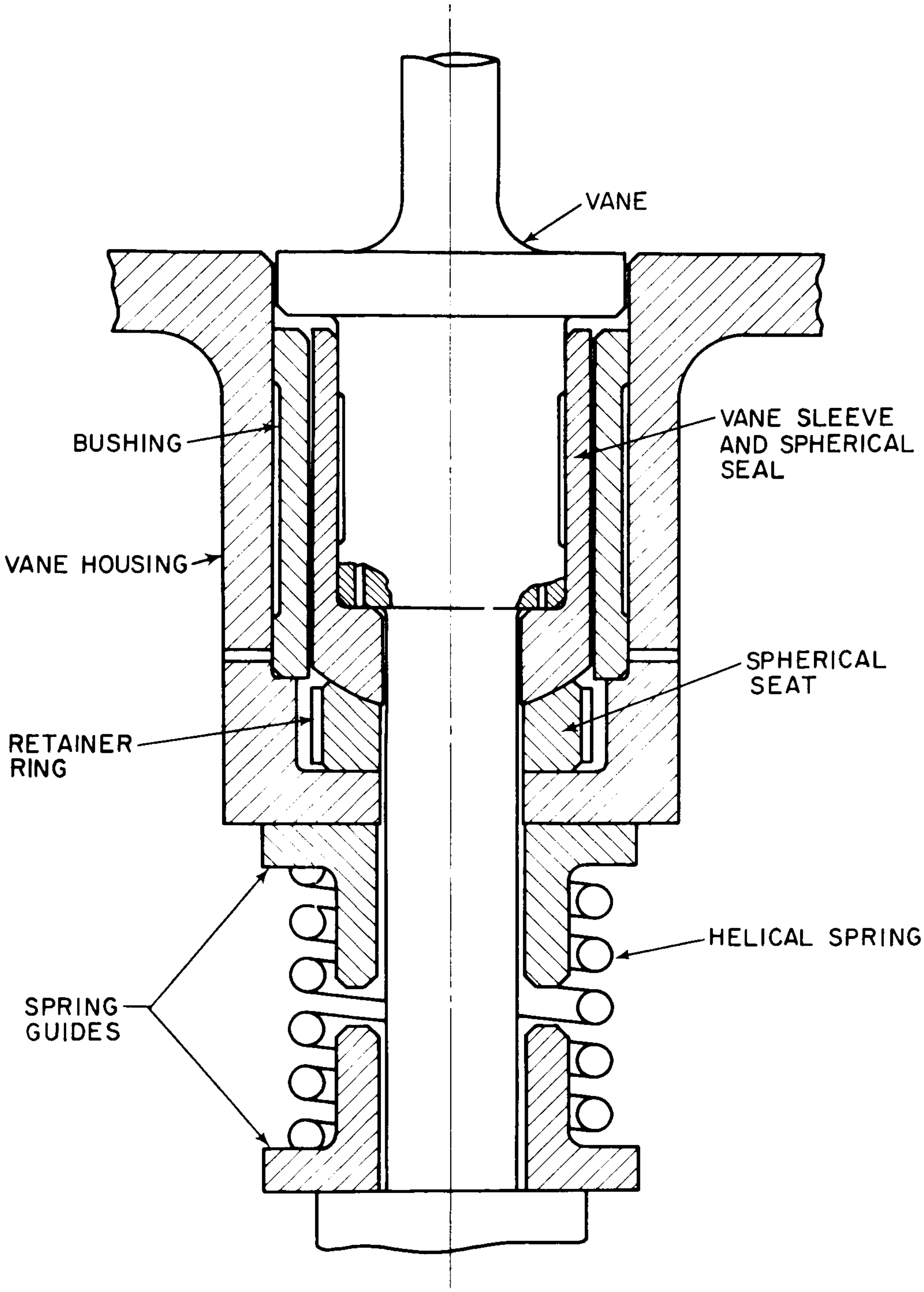


Figure 22 Spherical Seal Vane Pivot Seal

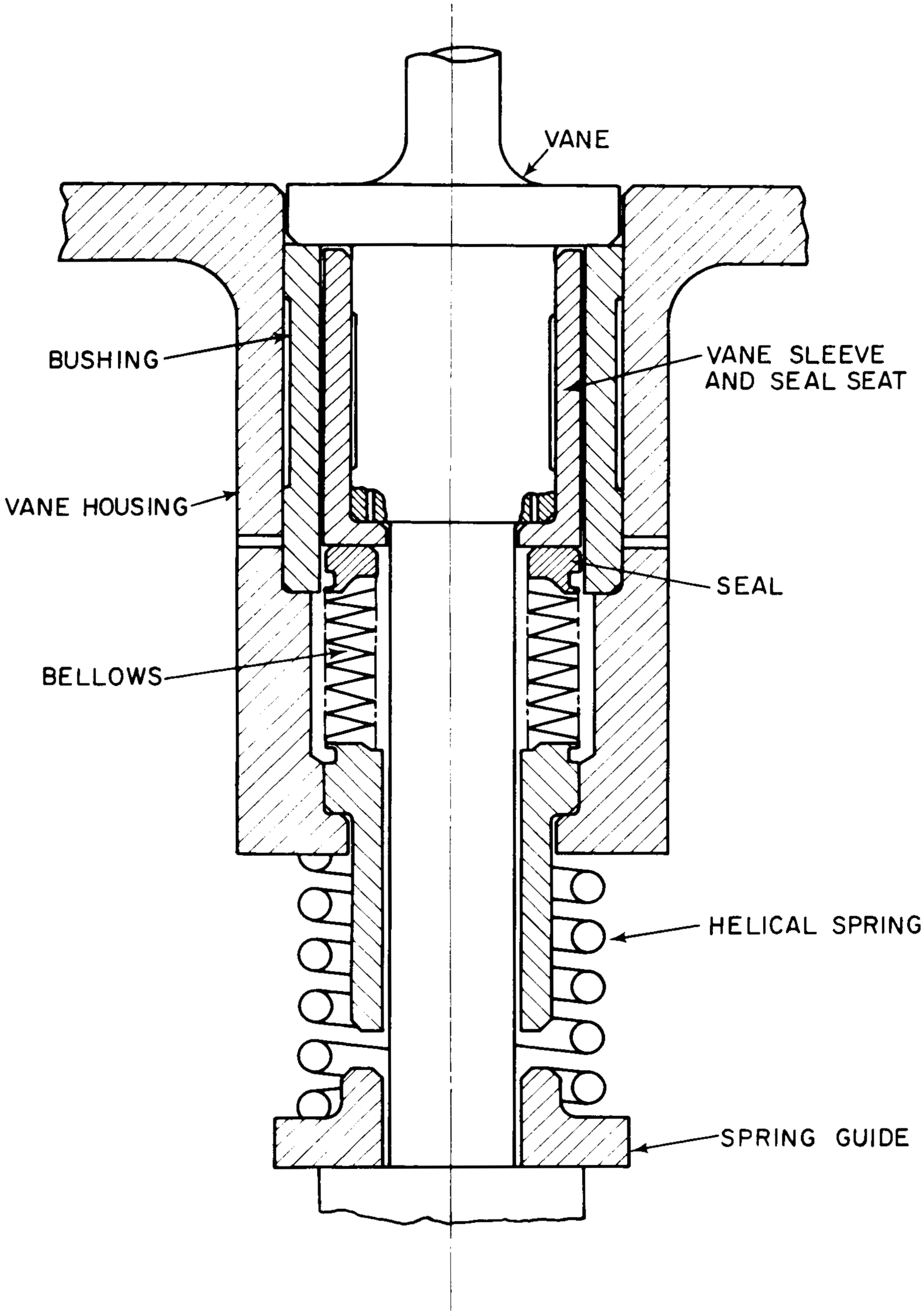
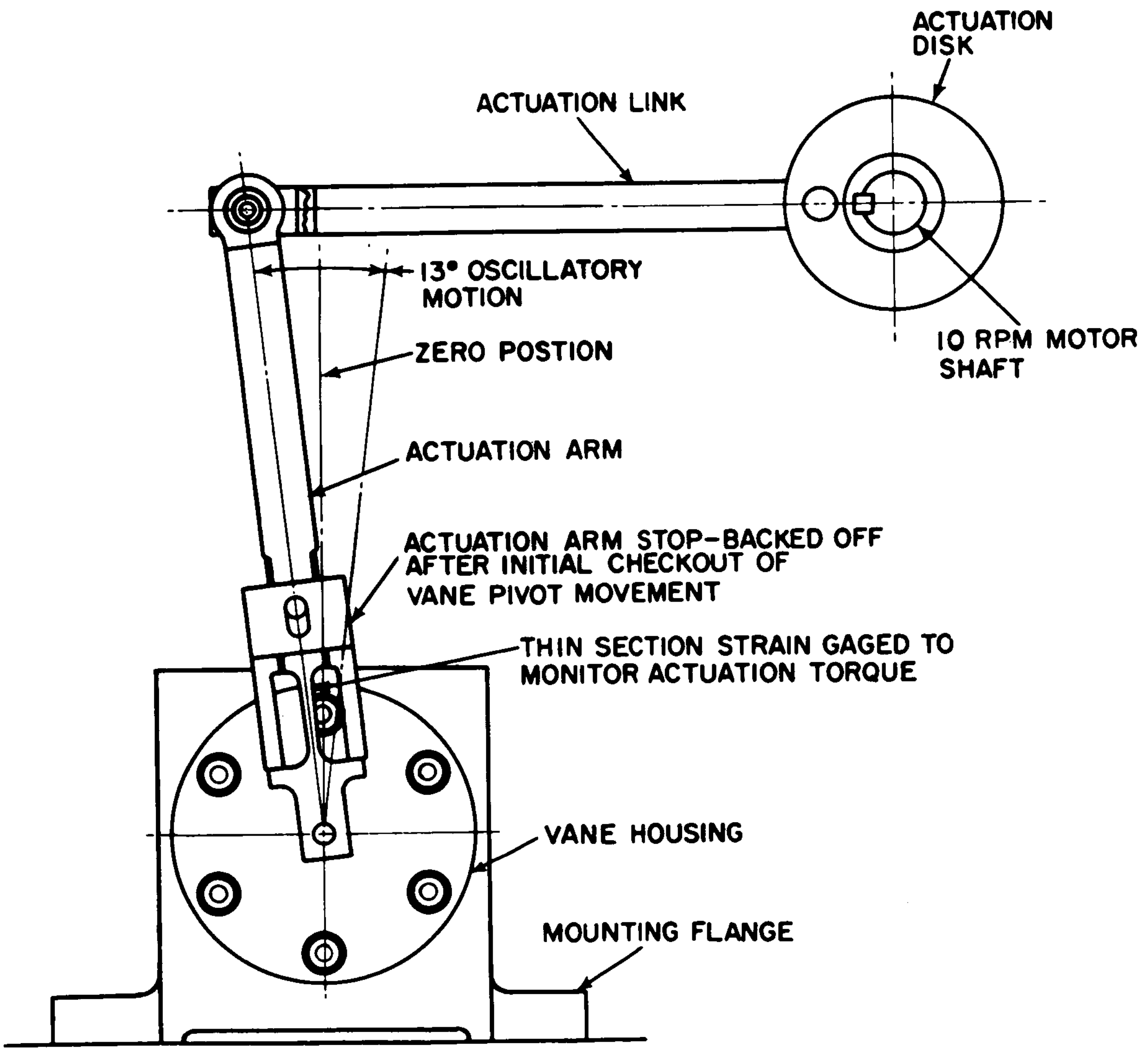


Figure 23 Single Bellows Vane Pivot Seal



VIEW IN DIRECTION C

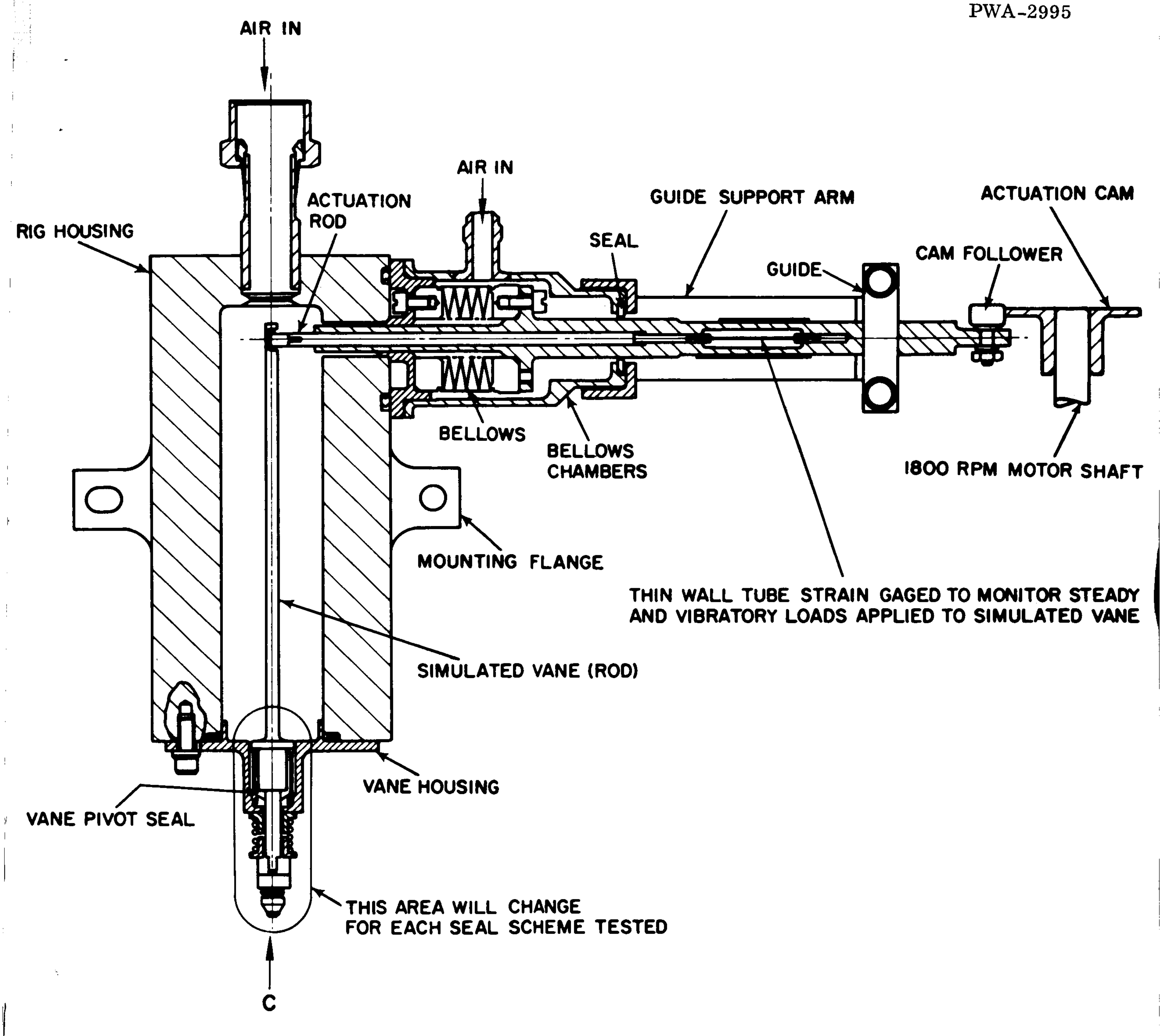


Figure 24 Vane Pivot Seal Test Rig

Spherical Seat Seal

Part	Material
Seal Seat	Aluminum Oxide (Coors AD-99C) Stellite 6 (AMS 5387) Carbon (Pure Carbon 56HT)
Seal	Chrome Carbide (Linde LC-1C) Tungsten Carbide (Linde LW-5)
Housing Bushing	Stellite 6 (AMS 5387)
Vane Sleeve	(Same as seal if initial testing indicates necessity)

Bellows Face Seal

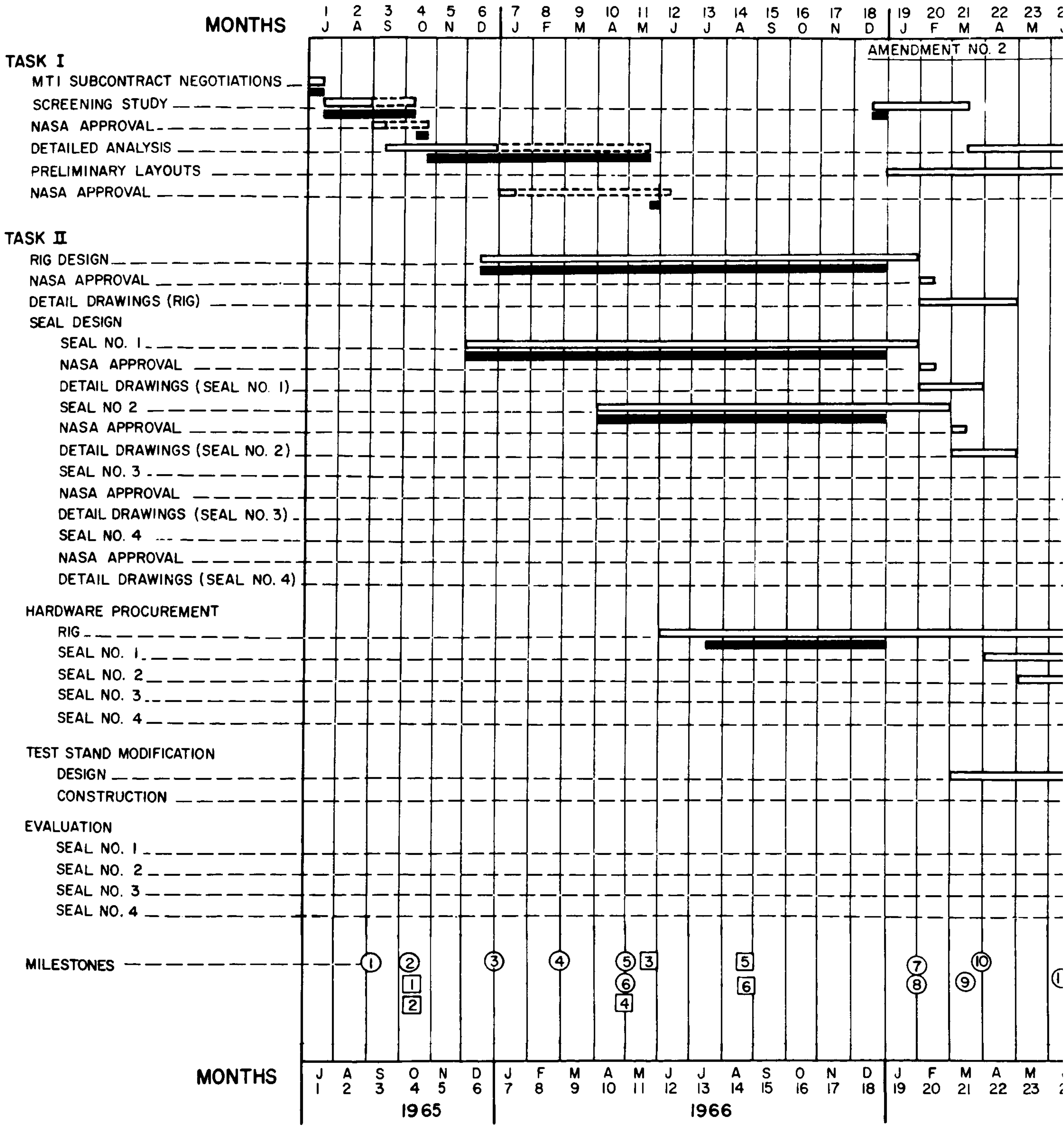
Part	Material
Seal Seat	Chrome Carbide (Linde LC-1C) Tungsten Carbide (Linde LW-5)

Part	Material
Seal	Chrome Carbide (Linde LC-1C) Aluminum Oxide (Linde LA-2) Tungsten Carbide (Linde LW-5)
Housing Bushing	Stellite 6 (AMS 5387)
Vane Sleeve	(Same as seal seat if initial testing indicates necessity)

COMPRESSOR SE

PROGRAM SCHEDULE

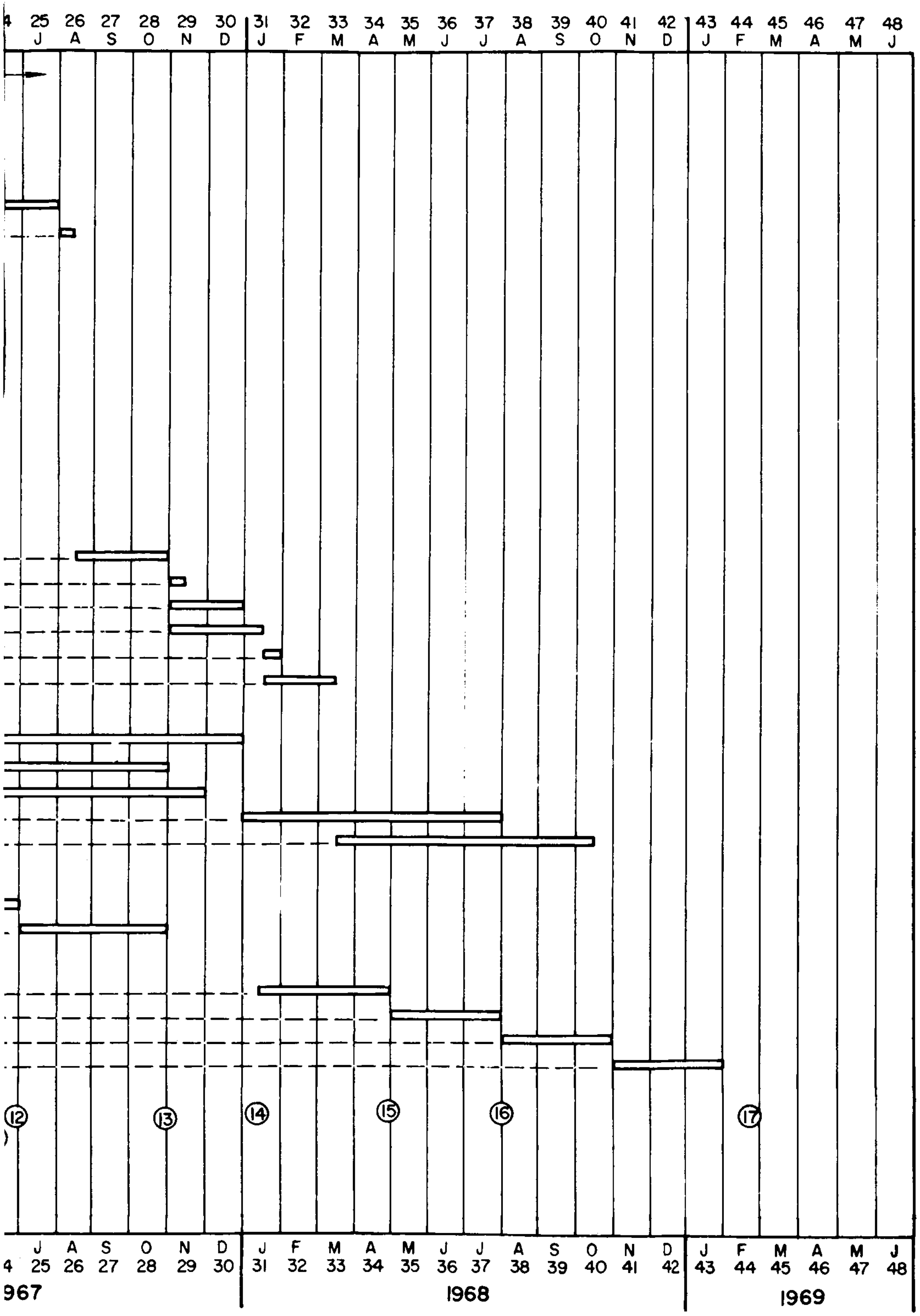
CONTRACT NAS3-76



AL DEVELOPMENT

AND MILESTONE CHART

5 – AMENDMENT NO. 2



MILESTONES

1. TASK I COMPLETE SCREENING STUDY
2. TASK III COMPLETE SCREENING STUDY
3. TASK I COMPLETE DETAILED ANALYSIS
4. TASK III COMPLETE DETAILED ANALYSIS
5 TASK IV COMPLETE RIG DESIGN
6 TASK IV COMPLETE SEAL DESIGN
7. TASK II COMPLETE RIG DESIGN
8. TASK II COMPLETE SEAL DESIGN (NO. 1)
9. TASK I COMPLETE SCREENING STUDY NO. 2
10. TASK IV INITIATE TEST
11. TASK I COMPLETE DETAILED ANALYSIS NO. 2
12. TASK IV COMPLETE EVALUATION OF ONE STATOR PIVOT SEAL
13. TASK II COMPLETE SEAL DESIGN (NO. 3)
14 TASK II INITIATE TEST
15. TASK II COMPLETE EVALUATION OF ONE COMPRESSOR END SEAL
16. TASK II COMPLETE EVALUATION OF ONE STATOR INTERSTAGE SEAL
17. SUBMIT SUMMARY REPORT FOR NASA APPROVAL

LEGEND

WORK PROJECTED

REVISED WORK PROJECTED

WORK ACCOMPLISHED

WORK ACHIEVED EARLY

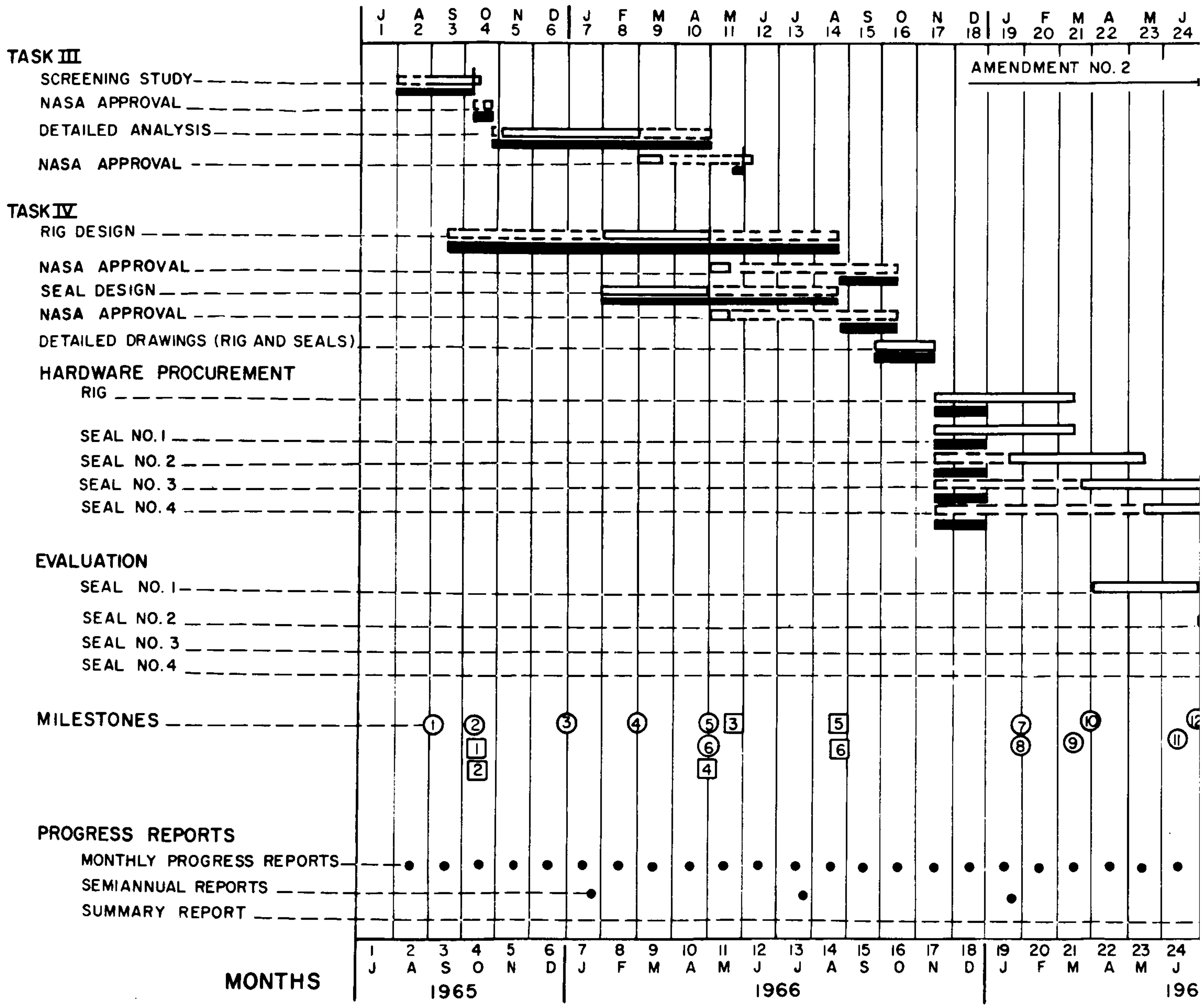
ORIGINAL MILESTONE

MILESTONE ATTAINED

COMPRESSOR SEA

PROGRAM SCHEDULE A

CONTRACT NAS3-7605



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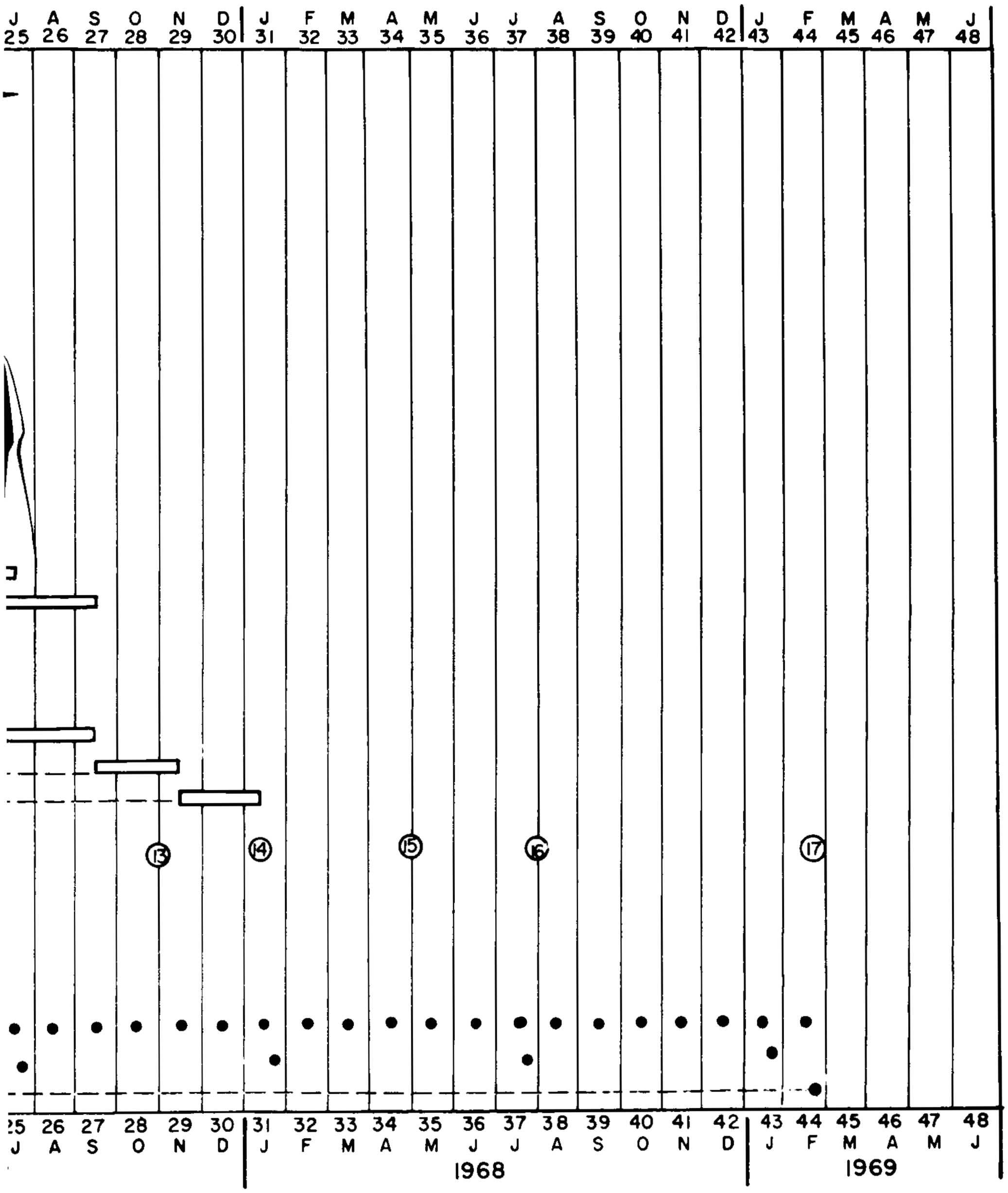
DEVELOPMENT

D MILESTONE CHART

- AMENDMENT NO. 2



MILESTONES

1. TASK I COMPLETE SCREENING STUDY
2. TASK III COMPLETE SCREENING STUDY
3. TASK I COMPLETE DETAILED ANALYSIS
4. TASK III COMPLETE DETAILED ANALYSIS
5. TASK IV COMPLETE RIG DESIGN
6. TASK IV COMPLETE SEAL DESIGN
7. TASK II COMPLETE RIG DESIGN
8. TASK II COMPLETE SEAL DESIGN (NO.1)
9. TASK I COMPLETE SCREENING STUDY NO. 2
10. TASK IV INITIATE TEST
11. TASK I COMPLETE DETAILED ANALYSIS NO. 2
12. TASK IV COMPLETE EVALUATION OF ONE STATOR PIVOT SEAL
13. TASK II COMPLETE SEAL DESIGN (NO.3)
14. TASK II INITIATE TEST
15. TASK II COMPLETE EVALUATION OF ONE COMPRESSOR END SEAL
16. TASK II COMPLETE EVALUATION OF ONE STATOR INTERSTAGE SEAL
17. SUBMIT SUMMARY REPORT FOR NASA APPROVAL

LEGEND

WORK PROJECTED

REVISED WORK PROJECTED

WORK ACCOMPLISHED

WORK ACHIEVED EARLY

ORIGINAL MILESTONE

MILESTONE ATTAINED

APPENDIX

FORTRAN LISTING FOR THE RAYLEIGH PAD SEAL

```
$EXECUTE       IBJOB
$IBJOB         ALTIO
$IBFTC MAIN    LIST,REF,DECK
C      RINO10 -REVISED BY H CHENG  1/28/66
C      CORRECTED BY C CHOW 12/6/66
C      MATRIX SIZE REDUCED FROM (17,33) TO  (15,20) H NORTHUP 7/4/66
C      PROGRAM TO SOLVE STEP  COMPRESS. BEAR. PROB. WITH FIXED
C      BOUNDARIES, LINES OF SYMMETRY,JOINTS IN ANY DIRECTION.
C      EQUATIONS ARE WRITTEN FOR PARALLEL FACES
C      ONLY. CLEARANCE ALLOWS ONE DEPRESSED AREA
C      ONLY.
C      KUE=0  REGULAR POINT OR CORNER OF DEPRESSED AREA OR LINE OF SUMM
C      KUE=1,2,3 KNOWN PRESSURE= PFIX(1,2,OR 3)
C      KUE=4  VERTICAL LINE OF STEP
C      KUE=5 HORIZONTAL LINE OF STEP
C      KUE=6 TOP JOINT
C      KUE=7 BOTTOM JOINT
C      KUE=8 LEFT JOINT
C      KUE=9 RIGHT JOINT
C      PROBLEM IS SOLVED COLUMNWISE. M (FIRST INDEX)
C      SHOULD BE SMALLER THAN N(SECOND  INDEX)
C      X IN I DIRECTION  (VERT. DOWN)
C      Y IN J DIRECTION (HOR. LEFT TO RIGHT)
C      PLAMX,PLAMY=X,Y,  COMPONENTS OF PLAM
C      (IH,JH),(IHH,JHH) ARE CORNERS OF STOP BOUNDARY
C      STEDE=STEP DEPTH. WHERE NO STEP    H=1
C      NDIG= NO OF DIGITS WANTED REPEATED TO TRUNCATE SOLUTION
C      LKOUNT IS THE MAXIMUM ALLOWABLE NUMBER OF ITERATIONS
C      IFLO= I COORDINATE OF THE LINE ACROSS WHICH Y-FLOW IS COMPUTED.
C      JFLO= J COORDINATE OF THE LINE ACROSS WHICH X-FLOW IS COMPUTED.
C      COE=CLEARANCE COEFFICIENTS. SEE HFUN, HXFUN, HYFUN.
C      QREP=.TRUE. PUT OUT P2 AFTEREACH ITERATION
C      PPOUT=.TRUE.  OUTPUT OF P2 AFTER CONVERGENCE
C      POUT=.TRUE.  WANTED OUTPUT OF P.AFTER CONVERGENCE
C      NEWKUE=.TRUE. IF NEW  KUE ARRAY IS READ IN
       DIMENSION PFIX(3),KUE(15,20),QFIX(3),H(15,20),PF(15,20),
```

```
 1Q(15,20),FF(15,20),F(15),A(15,15),B(15),C(20),E(15,15,21),
 1QSMA(15,15),G(15,21),R(15,15,20),S(15,20),QQ(20),PP(20),PX(20),
 1PY(20),D(15,15,21),QQQ(20),COE(10)
 1,XX(15),YY(20),PFX(15,20),PFY(15,20),HFX(15,20),HFY(15,20),F5(15,
 120)
  LOGICAL JOINT, QREP,PPOUT,POUT,NEWKUE
  NAMELIST/OUTPUT/QREP,PPOUT,POUT,NEWKUE/INPUT/M,N,PLAMX,
 1    PLAMY,YOX,IH,JH,IHH,JHH,STEDE,NDIG,PFIX,NCASE ,LKOUNT,IFLO,JFLO
 2,IFLOE,COE
1 FORMAT(1X70I1)
2 FORMAT(70I1)
3 FORMAT( 25H MATRIX IS SINGULAR AT J= I3,16H,CASE ABANDONED./1H1)
4 FORMAT( /                  10(1X,F11.7))
5 FORMAT( //18H CASE CONVERGES TO I3,14H DIGITS AFTER I3,11H ITERATI
 1ONS)
6 FORMAT(//23H FINAL RESULTS FOR CASE I5//13H FORCE/AREA =E14.7,
 1              64HC00. OF CENTER OF PRESSURE IN PERCENTAGE OF SIDE
 2DIMENSIONS = (E14.7,1H, E14.7,2H).)
7 FORMAT(46H0FLOW PER UNIT LENGTH IN X AT ENT. AND EXIT =(
 11PE14.7,2H, 1PE14.7,1H)24HFLOW PER U. L. IN Y = ( 1PE14.7,1H)/)
8 FORMAT(29H0FINAL PRESSURE DISTRIBUTION. //)
9 FORMAT(25H0FINAL P**2 DISTRIBUTION.  /)
11 FORMAT (6H1INPUT)
10 READ(5,INPUT)
   WRITE(6,11)
   WRITE(6,INPUT)
   READ(5,OUTPUT)
   WRITE(6,OUTPUT)
   IF(.NOT.NEWKUE)GO TO 35
   DO 20  I= 1,M
20 READ(5,2)(KUE(I,J),J=1,N)
   DO 30 I=1,M
   WRITE(6,1)(KUE(I,J),J=1,N)
   DO 30 J=1,N
30 KUE(I,J)=KUE(I,J)+1
35 KOUNT=0
```

```
      NN=N-1
      MM=M-1
      DO 40  K=1,3
40    QFIX(K)= PFIX(K)*PFIX(K)
      DX=1./FLOAT(MM)
      DY=YOX/FLOAT(NN)
      DO 41  I=1,M
41    XX(I)=FLOAT(I-1)*DX
       DO 42 J=1, N
42    YY(J)=FLOAT(J-1)*DY
      SA=1./DX
      SB=1./DY
      SAA= SA*SA
      SBB=SB*SB
      SC=SAA+SBB
      SD=-2.*SC
      SJ= 2.*SAA
      SE=PLAMX/(2.*DX)
       SEE=1./(2.*DX)
      SG=PLAMY/(2.*DY)
       SGG=1./(2.*DY)
      SH= 2.*SA
      SI=2.*SB
      SK=2.*SBB
      DO 50  I=1,M
      DO 50  J=1,N
50    H(I,J)=HFUN(XX(I),YY(J),COE)
      DO 60  I=IH,IHH
      DO 60  J=JH,JHH
60    H(I,J)=H(I,J)+STEDE
      DO 70  I=1,M
      DO 70  J=1,N
      Q(I,J)=  1
       HFX(I,J)=3.*HXFUN(XX(I),YY(J),COE)/H(I,J)
       HFY(I,J)=3.*HYFUN(XX(I),YY(J),COE)/H(I,J)
      ZZZ=-1.0/(H(I,J)*H(I,J))
```

```
      F5(I,J)=(2./3.)*ZZZ*(HFX(I,J)*PLAMX+HFY(I,J)*PLAMY)
      PFX(I,J)=PLAMX*ZZZ
   70 PFY(I,J)=PLAMY*ZZZ
      JOINT=.FALSE.
   80 DO 90     I=2,MM
      IF(KUE(I,1).EQ.9.OR.KUE(I,1).EQ.10) JOINT=.TRUE.
   90 CONTINUE
  100 DO 130 I=1,M
      G(I,1)=0.
      DO 110  K=1,M
  110 E(I,K,1)=0.
      IF(.NOT.JOINT) GO TO 130
      DO 120  K=1,M
      D(I,K,1)=0.
      IF( I.EQ.K)D(I,K,1)=1.0
  120 CONTINUE
  130 CONTINUE
      DO 370 J=1,N
      DO 310 I=1,M
      FF(I,J)=1.0/SQRT(ABS(Q(I,J)))
      F(I)=0.
      KU= KUE (I,J)
      GO TO(140,210,210,210,230,250,270,270,140,140),KU
  140 SGGG=SGG *(FF(I,J)*PFY(I,J)+HFY(I,J))
      C(I)=SBB+SGGG
      B(I)=SBB-SGGG
      DO 150 K=1,M
      A(I,K)=0.0
      IF(K.EQ.I)A(I,K)=SD+F5(I,J)*FF(I,J)
  150 CONTINUE
      IF(JOINT) GO TO 180
      IF(J.NE.1) GO TO 160
      B(I)=0.
      C(I)=SK
  160 IF (J.NE.N) GO TO 170
      C(I)=0.
```

```
      B(I)=SK
  170 IF(I.EQ.1) GO TO 190
      IF(I.EQ.M) GO TO 200
  180 SR=SEE*(FF(I,J)*PFX(I,J)+HFX(I,J))
      A(I,I+1)= SAA+SR
      A(I,I-1)=SAA-SR
      GO TO 310
  190 A(I,I+1)=SJ
      GO TO 310
  200 A(I,I-1)=SJ
      GO TO 310
  210 KKU=KUE(I,J)-1
      B(I)=0.
      C(I)=0.
      F(I)=QFIX(KKU)
      DO 220 K=1,M
      A(I,K)=0.
      IF(I.EQ.K) A(I,K)=1.0
  220 CONTINUE
      GO TO 310
  230 HPLUS=H(I,J)
      HMINUS=H(I,J)
      IF (J.EQ.JH)  HMINUS=HMINUS-STEDE
      IF (J.EQ.JHH) HPLUS=HPLUS-STEDE
      HHH=HPLUS**3
      HH=HMINUS**3
C     CORRECTED THE FLOW CONDITION AT THE STEP, 12/6/66
       S8=.5*SB
      B(I)= -S8 * HH
      C(I)= -S8 * HHH
      DO 240  K=1,M
  240 A(I,K)= 0.
      A(I,I)=S8*(HH+HHH)+PLAMY*( HPLUS-HMINUS)*FF(I,J)
      GO TO 310
  250 HPLUS=H(I,J)
      HMINUS=H(I,J)
```

```
      IF (I.EQ.IH)  HMINUS=HMINUS-STEDE
      IF (I.EQ.IHH) HPLUS=HPLUS-STEDE
      HH=HMINUS**3
      HHH=HPLUS**3
      B(I)=0.
      C(I)=0.
        D9=2.*DX
      DO 260    K=1,M
  260 A(I,K)=0.0
      A(I,I)=(HH+HHH)/D9-PLAMX*(HMINUS-HPLUS)*FF(I,J)
      A(I,I+1)=-HHH/D9
      A(I,I-1)=-HH/D9
      GO TO 310
  270 SGGG=SGG*(FF(I,J)*PFY(I,J)+HFY(I,J))
      B(I)=SBB-SGGG
      C(I)=SBB+SGGG
      DO 280  K=1,M
      A(I,K)=0.0
      IF(K.EQ.I) A(I,K)=SD +F5(I,J)*FF(I,J)
  280 CONTINUE
      SR=SEE*(FF(I,J)*PFX(I,J)+HFX(I,J))
      IF( KU. EQ.8) GO TO 290
      IF( KU.EQ.7) GO TO 300
      GO TO 310
  290 A(I,1)=SAA+SR
      A(I,I-1)=SAA-SR
      GO TO 310
  300 A(I,M)=SAA-SR
      A(I,I+1)=SAA+SR
       GO TO 310
  310 CONTINUE
      DO 320    I=1,M
      DO 320    K=1,M
```

```
320 QSMA(I,K)=A(I,K) + B(I)*E(I,K,J)
    CALL MATINV(QSMA,M,BB,0,DET,ID)
    GO TO (340,330),ID
340 DO 360   I=1,M
    G(I,J+1)=0.
    DO 360   K=1,M
    G(I,J+1)=G(I,J+1)+QSMA(I,K)*(F(K)-B(K)*G(K,J))
    E(I,K,J+1)=-QSMA(I,K)*C(K)
    IF(.NOT.JOINT) GO TO 360
    DUM=0.0
    DO     350 KK=1,M
350 DUM=DUM-QSMA(I,KK)*B(KK)*D(KK,K,J)
    D(I,K,J+1)=DUM
360 CONTINUE
370 CONTINUE
    DMA=0.0
    IF(JOINT) GO TO 410
    DO 380 I=1,M
    DMA=AMAX1(DMA,ABS(Q(I,N)-G(I,N+1)))
380 Q(I,N)=G(I,N+1)
    DO 400 JJ=2,N
    J=N+2-JJ
    DO     400 I=1,M
    DUM=0.0
    DO 390     K=1,M
390 DUM=DUM+E(I,K,J)*Q(K,J)
    DUM=DUM+G(I,J)
    DMA=AMAX1(DMA,ABS(DUM-Q(I,J-1)))
400 Q(I,J-1)=DUM
    GO TO 560
410 DO 420    I=   1,M
    DO 420    K=1,M
    QSMA(I,K)=   -D(I,K,N+1)
    IF(I.EQ.K)QSMA(I,K)=QSMA(I,K)+1.0
420 CONTINUE
    CALL MATINV(QSMA,M,BB,0,DET,ID)
```

```
      GO TO (430,330),ID
  330 WRITE(NW,3) J
      GO TO 10
  430 DO 460 I=1,M
      DU= 0.0
      DO 450  K=1,M
      DUM= 0.0
      DO 440 KK=1,M
  440 DUM=DUM+QSMA(I,KK)*E(KK,K  ,N+1)
      R(I,K,N)=DUM
  450 DU=DU+QSMA(I,K)*G(K,N+1)
  460 S(I,N)=DU
      DO 490 JJ=2,N
      J= N+2-JJ
      DO 490   I=1,M
      DU=0.0
      DO 480 K=1,M
      DUM=0.0
      DO 470 KK=1,M
  470 DUM= D(I,KK,J)*R(KK,K,N)+E(I,KK,J)*R(KK,K,J)+DUM
      R(I,K,J-1)=DUM
  480 DU=DU+D(I,K,J)*S(K,N)+E(I,K,J)*S(K,J)
  490 S(I,J-1)=DU+G(I,J)
      DMA=0.0
      DO 500  I=1,M
      DO 500   K=1,M
      QSMA(I,K)=-R(I,K,1)
      IF(I.EQ.K)QSMA(I,K)= QSMA(I,K)+1.0
  500 CONTINUE
      CALL MATINV(QSMA,M,BB,0,DET,ID)
      GO TO(510,330),ID
  510 DO 530  I=1,M
      DU=0.0
      DO 520  K=1,M
  520 DU=DU+QSMA(I,K)*S(K,1)
      DMA=AMAX1(DMA,ABS(DU-Q(I,1)))
```

```
530 Q(I,1)=DU
    DO 550 J=2,N
    DO 550 I=1,M
    DU=0.0
    DO 540  K=1,M
540 DU=DU+R(I,K,J)*Q(K,1)
    DU=DU+S(I,J)
    DMA= AMAX1(DMA,ABS(DU-Q(I,J)))
550 Q(I,J)=DU
560 IF(QREP) WRITE(6,4)Q
    KOUNT=KOUNT+1
    IF(KOUNT.GE.LKOUNT)GO TO 561
    IF(DMA   .GT. 10.0**FLOAT(-NDIG)                    )GO TO 100
561 WRITE(6,5)NDIG,KOUNT
    WRITE(6,11)
    IF(PPOUT)WRITE(6,9)
    DO 575 I=1,M
    IF(PPOUT)WRITE(6,4)(Q(I,J),J=1,N)
    DO 570  J=1,N
570 Q(I,J)=SQRT(ABS(Q(I,J)))
575 CONTINUE
    IF(POUT)WRITE(6,8)
    DO 576 I=1,M
    IF(POUT)WRITE(6,4)(Q  (I,J),J=1,N)
576 CONTINUE
    DO 590 I=1,M
    X= DX*FLOAT(I-1)
    DO 580  J=1,N
580 QQQ(J)=Q(I,J)-1.0
    PP(I)= SUM(QQQ,N,DY)
590 PX(I)= PP(I)*X
    DO 610  J=1,N
    Y= FLOAT( J-1)*DY
    DO 600 I=1,M
600 QQQ(I)=Q(I,J)-1.0
610 PY(J)=SUM(QQQ,M,DX)*Y
```

```
      FP= SUM(PP,M,DX)
      FX= SUM(PX,M,DX)
      XF =FX/FP
      FY= SUM(PY,N,DY)
      YF=FY/FP/YOX
      FP=FP/YOX
      WRITE(6,6) NCASE,FP,XF,YF
      DO 620 I=1,M
  620 PP(I)=Q(I,JFLO)*H(I,JFLO)*(-PLAMY+H(I,JFLO)**2
     1*(Q(I,JFLO+1)-Q(I,JFLO-1))/(2.*DY))
      FLOY=SUM(PP,M,DX)
      NFLO=1
      IFLT=IFLO
  625 DO 630 J=1,N
  630 PP(J)=Q(IFLT,J)*H(IFLT,J)*(-PLAMX+H(IFLT,J)**2
     1*(Q(IFLT+1,J)-Q(IFLT-1,J))/(2.*DX))
      FLOT=SUM(PP,N,DY)/YOX
      GO TO (635,636),NFLO
  635 NFLO=2
      FLOX=FLOT
      IFLT=IFLOE
      GO TO 625
  636 CONTINUE
      WRITE(6,7) FLOT,FLOX,FLOY
      GO TO 10
      END
$IBFTC HFUN     LIST,REF,DECK
       FUNCTION HFUN (XX,YY,COE)
C     FUNCTION TO EVALUATE CLEARANCE H FROM X,Y AND COEFFICIENTS COE
      DIMENSION COE(10)
      X=XX-COE(1)
      Y=YY-COE(2)
      HFUN=COE(3)+COE(4)*X+COE(5)*Y+COE(6)*X*X+
     1COE(7)*Y*Y+COE(8)*X*Y+COE(9)*X*X*X+
     2COE(10)*Y*Y*Y
      RETURN
```

```
      END
$IBFTC HXFUN    LIST,REF,DECK
      FUNCTION HXFUN(XX,YY,COE)
C     FUNCTION TO EVALUATE X-DERIVATIVE OF H FROM X,Y AND COEFFICIENTS C
      DIMENSION COE(10)
      X=XX-COE(1)
      Y=YY-COE(2)
      HXFUN=COE(4)+2.*COE(6)*X+COE(8)*Y+
     13.*COE(9)*X*X
      RETURN
      END
$IBFTC HYFUN    LIST,REF,DECK
      FUNCTION HYFUN(XX,YY,COE)
C     FUNCTION TO EVALUATE Y-DERIVATIVE OF H FROM X,Y AND COEFFICIENTS C
      DIMENSION COE(10)
      X=XX-COE(1)
      Y=YY-COE(2)
      HYFUN=COE(5)+2.*COE(7)*Y+COE(8)*X+3.*COE(10)*Y*Y
      RETURN
      END
$IBFTC SUM      LIST,REF,DECK
      FUNCTION SUM(P,M,DX)
      DIMENSION P(20)
      K=2
      KK=M-1
      KKK=2
      SUM=0.0
   10 DO 20 I=K,KK,KKK
   20 SUM=SUM+P(I)
      GO TO (30,40,50),K
   30  SUM=SUM*DX/3.0
      RETURN
   40 K=3
   45 SUM=2.0*SUM
      GO TO 10
   50 K=1
```

```
      KK=M                                                              53
      KKK=M-1                                                           54
      GO TO 45                                                          55
      END                                                               56
$IBFTC MATINV  LIST,REF,DECK
C     MATRIX INVERSION WITH ACCOMPANYING SOLUTION OF LINEAR EQUATIONS   59
C     NOVEMBER 1692   S GOOD  DAVID TAYLOR MODEL BASIN   AM MAT1        60
      SUBROUTINE MATINV(A,N1,B,M1,DETERM,ID)                            62
C                                                                       63
C     GENERAL FORM OF DIMENSION STATEMENT                               64
C                                                                       65
      DIMENSION A(15,15),B(15,1),INDEX(15,3)                            66
      EQUIVALENCE (IROW,JROW), (ICOLUM,JCOLUM), (AMAX, T, SWAP)         67
C                                                                       68
C     INITIALIZATION                                                    69
C                                                                       70
      M=M1                                                              71
      N=N1                                                              72
   10 DETERM=0.C                                                        73
   15 DO 20 J=1,N                                                       74
   20 INDEX(J,3) = 0                                                    75
   30 DO 550 I=1,N                                                      76
C                                                                       77
C     SEARCH FOR PIVOT ELEMENT                                          78
C                                                                       79
   40 AMAX=0.0                                                          80
   45 DO 105 J=1,N                                                      81
      IF(INDEX(J,3)-1) 60, 105, 60                                      82
   60 DO 100 K=1,N                                                      83
      IF(INDEX(K,3)-1) 80, 100, 715                                     84
   80 IF (      AMAX -ABS (A(J,K))) 85, 100, 100                        85
   85 IROW=J                                                            86
   90 ICOLUM=K                                                          87
      AMAX = ABS (A(J,K))                                               88
  100 CONTINUE                                                          89
  105 CONTINUE                                                          90
```

```
      INDEX(ICOLUM,3) = INDEX(ICOLUM,3) +1
  260 INDEX(I,1)=IROW
  270 INDEX(I,2)=ICOLUM
C
C     INTERCHANGE ROWS TO PUT PIVOT ELEMENT ON DIAGONAL
C
  130 IF (IROW-ICOLUM) 140, 310, 140
  140 DETERM=-DETERM
  150 DO 200 L=1,N
  160 SWAP=A(IROW,L)
  170 A(IROW,L)=A(ICOLUM,L)
  200 A(ICOLUM,L)=SWAP
      IF(M) 310, 310, 210
  210 DO 250 L=1, M
  220 SWAP=B(IROW,L)
  230 B(IROW,L)=B(ICOLUM,L)
  250 B(ICOLUM,L)=SWAP
C
C     DIVIDE PIVOT ROW BY PIVOT ELEMENT
C
 310  PIVOT   =A(ICOLUM,ICOLUM)
      DETERM=DETERM*PIVOT
  330 A(ICOLUM,ICOLUM)=1.0
  340 DO 350 L=1,N
  350 A(ICOLUM,L)=A(ICOLUM,L)/PIVOT
  355 IF(M) 380, 380, 360
  360 DO 370 L=1,M
  370 B(ICOLUM,L)=B(ICOLUM,L)/PIVOT
C
C     REDUCE NON-PIVOT ROWS
C
  380 DO 550 L1=1,N
  390 IF(L1-ICOLUM) 400, 550, 400
  400 T=A(L1,ICOLUM)
  420 A(L1,ICOLUM)=0.0
  430 DO 450 L=1,N
```

```
  450 A(L1,L)=A(L1,L)-A(ICOLUM,L)*T
  455 IF(M) 550, 550, 460
  460 DO 500 L=1,M
  500 B(L1,L)=B(L1,L)-B(ICOLUM,L)*T
  550 CONTINUE
C
C     INTERCHANGE COLUMNS
C
  600 DO 710 I=1,N
  610 L=N+1-I
  620 IF (INDEX(L,1)-INDEX(L,2)) 630, 710, 630
  630 JROW=INDEX(L,1)
  640 JCOLUM=INDEX(L,2)
  650 DO 705 K=1,N
  660 SWAP=A(K,JROW)
  670 A(K,JROW)=A(K,JCOLUM)
  700 A(K,JCOLUM)=SWAP
  705 CONTINUE
  710 CONTINUE
      DO 730 K = 1,N
      IF(INDEX(K,3) -1) 715,720,715
  715 ID =2
      GO TO 740
  720 CONTINUE
  730 CONTINUE
      ID=1
  740 RETURN
C     LAST CARD OF PROGRAM
      END
```

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    20th Street and Indiana Avenue
    Philadelphia 32, Pennsylvania
    Attention: Dr. P. C. Stein

18. Wright Aeronautical Division
    Curtiss-Wright Corporation
    333 West 1st Street
    Dayton 2, Ohio
    Attention: S. Lombardo

19. General Electric Company
    Advanced Engine and Technology Department
    Cincinnati, Ohio 45215
    Attention: L. B. Venable
    G. J. Wile
    C. C. Moore H-25

20. Huyck Metals Company
    P. O. Box 30
    45 Woodmont Road
    Milford, Connecticut
    Attention: J. I. Fisher

21. Aerojet-General Corporation
    20545 Center Ridge Road
    Cleveland, Ohio 44116
    Attention: W. L. Snapp

22. Lycoming Division
    Avco Corporation
    Stratford, Connecticut
    Attention: R. Cuny

23. Battelle Memorial Institute
    505 King Avenue
    Columbus 1, Ohio
    Attention: C. M. Allen

24. Bendix Corporation
    Fisher Building
    Detroit 2, Michigan
    Attention: R. H. Isaccs

25. Boeing Aircraft Company
    224 N. Wilkinson
    Dayton, Ohio 45402
    Attention: H. W. Walker

26. Douglas Aircraft Company
    Holiday Office Center
    16501 Brookpark Road
    Cleveland, Ohio 44135
    Attention: J. J. Pakiz

27. General Dynamics Corporation
    16501 Brookpark Road
    Cleveland, Ohio 44135
    Attention: George Vila

28. General Motors Corporation
    Allison Division
    Plant #8
    Indianapolis, Indiana
    Attention: E. H. Deckman

29. Lockheed Aircraft Company
    16501 Brookpark Road
    Cleveland, Ohio 44135
    Attention: Mr. L. Kelly

30. Martin Company
    16501 Brookpark Road
    Cleveland, Ohio 44135
    Attention: Z. G. Horvath

31. North American Aviation
    16501 Brookpark Road
    Cleveland, Ohio 44135
    Attention: George Bremer

32. Fairchild-Hiller Corporation
    Republic Aviation Division
    Farmingdale, Long Island
    New York 11735
    Attention: D. Schroeder

33. Westinghouse Electric Corporation
    55 Public Square
    Cleveland, Ohio 44113
    Attention: Lynn Powers

34. I. I. T. Research Foundation
    10 West 35 Street
    Chicago, Illinois 60616
    Attention: Dr. Strohmeler

35. Pesco Products Division
    Borg-Warner Corporation
    24700 N. Niles
    Bedford, Ohio

36. Stanford Research Institute
    Menlo Park, California
    Attention: R. C. Fey

37. Franklin Institute Laboratories
    20th and Parkway
    Philadelphia 3, Pennsylvania
    Attention: J. V. Carlson

38. Industrial Tectonics
    Box 401
    Hicksville, New York 11801
    Attention: J. Cherubin

39. Sealol Incorporated
    P. O. Box 2158
    Providence 5, Rhode Island
    Attention: Justus Stevens

40. Continental Aviation and Engineering
    12700 Kercheval
    Detroit 15, Michigan
    Attention: A. J. Fallman

41. Northrop Corporation
    1730 K. Street, N. W.
    Suite 903-5
    Washington 6, D. C.
    Attention: S. W. Fowler, Jr.

42. Chicago Rawhide Manufacturing Company
    1311 Elston Avenue
    Chicago, Illinois
    Attention: R. Blair

43. Midwest Research Institute
425 Volker Blvd.
Kansas City 10, Missouri
Attention: V. Hopkins

44. Southwest Research Institute
San Antonio, Texas
Attention: P. M. Ku

45. E. I. DuPont de Nemours and Company
1007 Market Street
Wilmington 98, Delaware
Attention: A. J. Cheney
R. J. Laux

46. Fairchild Engine and Airplane Corporation
Stratos Division
Bay Shore, New York

47. Borg-Warner Corporation
Roy C. Ingersoll Research Center
Wolf and Algonquin Roads
Des Plaines, Illinois

48. U. S. Naval Air Material Center
Aeronautical Engine Laboratory
Philadelphia 12, Pennsylvania
Attention: A. L. Lockwood

49. Department of the Navy
Bureau of Naval Weapons
Washington, D. C.
Attention: A. B. Nehman, RAAE-3
C. C. Singleterry, RAPP-4

50. Department of the Navy
Bureau of Ships
Washington, D. C.
Attention: Harry King, Code 634-A

51. SKF Industries, Incorporated
1100 First Avenue
King of Prussia, Pennsylvania
Attention: L. B. Sibley

52. Crane Packing Company
6400 W. Oakton Street
Norton Grove, Illinois
Attention: Harry Tankus

53. B. F. Goodrich Company
Aerospace and Defense Products Division
Troy, Ohio
Attention: L. S. Blaikowski

54. The University of Tennessee
Department of Mechanical and Aerospace Engineering
Knoxville, Tennessee
Attention: Professor W. K. Stair

55. Hughes Aircraft Company
International Airport Station
P. O. Box 90515
Los Angeles 9, California

56. U. S. Navy Marine Engineering Lab.
Friction and Wear Division
Annapolis, Maryland
Attention: R. B. Snapp

57. Metal Bellows Corporation
20977 Knapp Street
Chatsworth, California
Attention: Sal Artino

58. Rocketdyne
6633 Canoga Avenue
Canoga Park, California
Attention: M. Butner

59. Carbon Products Division of Union Carbide Corporation
270 Park Avenue
New York 17, New York
Attention: J. Curean

60. Garlock, Incorporated
Palmyra, New York 14522
Attention: E. W. Fisher

61. Chemicals Division of Union Carbide Corp.
    Technical Service Lab.
    P. O. Box 65
    Tarrytown, New York
    Attention: J. C. Haaga

62. Durametallic Corporation
    Kalamazoo, Michigan
    Attention: H. Hummer

63. Morganite, Incorporated
    33-02 48th Avenue
    L. I. C. 1, New York
    Attention: S. A. Rokaw

64. United States Graphite Company
    1621 Holland
    Saginaw, Michigan
    Attention: F. F. Ruhl

65. Cartiseal Corporation
    3515 West Touhy
    Lincolnwood, Illinois
    Attention: R. Voitik

66. Department of the Army
    U. S. Army Aviation Material Laboratory
    Fort Eustis, Virginia 23604
    Attention: John W. White, Chief
    Propulsion Division

67. AVCOM
    AMSAVEGTT
    Mart Building
    405 South 12th Street
    St. Louis, Missouri 63100
    Attention: E. England

455-10

Pratt & Whitney Aircraft DIVISION OF UNITED AIRCRAFT CORPORATION

20 March 1967

In reply please refer to:
RPS:RTD:ah - Eng. 2J

To:	National Aeronautics and Space Administration Lewis Research Center Air-Breathing Engine Procurement Section 21000 Brookpark Road Cleveland, Ohio 44135
Attention:	Mr. John H. DeFord, Contracting Officer
Subject:	Semiannual Progress Report No. 3 on Development of Compressor End Seals, Stator Interstage Seals, and Stator Pivot Seals in Advanced Air Breathing Propulsion Systems, PWA-2995
Reference:	Contract NAS3-7605
Enclosure:	One Copy of Subject Report

The subject report is submitted in compliance with the terms of the referenced contract.

UNITED AIRCRAFT CORPORATION
Pratt & Whitney Aircraft

R. P. Shevchenko
Senior Project Engineer